Chapter 2
Selection of Main Dimensions and Calculation of Basic Ship Design Values

Abstract This chapter deals with the determination of the main ship dimensions (length, beam, draft, side depth), following the estimation of the ship's displacement and the selection of other basic ship design quantities and hull form characteristics (hull form coefficients, powering, weight components, stability and trim, freeboard, load line), as required in the first phase of ship design, that is, the *Concept Design*. The various effects of specific selections of ship's main dimensions etc. on the ship's hydrodynamic performance, stability and trim, structural weight and construction cost, utilization of spaces, and transport economy are elaborated. The selection procedure is supported by statistical data and empirical design formulas, design tables and diagrams allowing direct applications to individual ship designs. Additional reference material is given in Appendix A.

2.1 Preliminary Estimation of Displacement

For deadweight carriers (Sect. 1.3.7.1), which are characterized by the carriage of relatively heavy cargos (low cargo Stowage Factor (SF) and low Ship Capacity Factor), but also for every category/type of ship with sufficient comparative data from similar ships on vessel's displacement, the preliminary design starts with the estimation of ship's displacement weight Δ.

For deadweight carriers, it is possible to estimate Δ for a given deadweight DWT, for instance, as the DWT is one of shipowner's main requirements.

Typical ways of estimating Δ are the following:

a. Using DWT/Δ ratios for various types of ships (see Table 2.1);
b. Using semiempirical mathematical formulae from statistics, regression analyses of data of similar vessels (see, for example analysis of technical database for various types of ships, such as the database of IHS Fairplay (IHS WSE 2011, former Lloyds Register of Shipping), and data from regression analyses studies of

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Table 2.1 Typical sizes and percentages of weight groups for main merchant ship types (compilation of data from Strohbusch (1971), Schneekluth (1985), updated by Papanikolaou using IHS Fairplay World Shipping Encyclopedia, v. 12.01, 2011)

	1	**2**	3	4	5	6
Ship type	Limits		DWT/Δ (%)	W_{ST}/W_L (%)	W_{OT}/W_L (%)	W_M/W_L (%)
	Lower	Upper				
General cargo ships (t DWT)	5,000	15,000	65–80	55–64	19–33	11–22
Coasters, cargo ships (GRT)	499	999	70–75	57–62	30–33	9–12
Bulk carriers[a] (t DWT)	20,000	50,000	74–85	68–79	10–17	12–16
	50,000	200,000	80–87	78–85	6–13	8–14
Tankers[b] (t DWT)	25,000	120,000	78–86	73–83	5–12	11–16
	200,000	500,000	83–88	75–88	9–13	9–16
Containerships (t DWT)	10,000	15,000	65–74	58–71	15–20	9–22
	15,000	165,000[c]	65–76	62–72	14–20	15–18
Ro-Ro (cargo) (t DWT)	$L \cong 80$ m	16,000 t DWT	50–60	68–78	12–19	10–20
Reefers[d] (ft^3) of net ref. vol.	300,000	500,000	45–55	51–62	21–28	15–26
Passenger Ro-Ro/ferries/ RoPax	$L \cong 85$ m	$L \cong 120$ m	16–33	56–66	23–28	11–18
Large passenger ships (cruise ships)	$L \cong 200$ m	$L \cong 360$[e] m	23–34	52–56	30–34	15–20
Small passenger ships	$L \cong 50$ m	$L \cong 120$ m	15–25	50–52	28–31	20–29
Stern Trawlers	$L \approx 44$ m	$L \cong 82$ m	30–58	42–46	36–40	15–20
Tugboats	$P_B \cong 500$ KW	3,000 KW	20–40	42–56	17–21	38–43
River ships (towed)	$L \cong 32$ m	$L \cong 35$ m	22–27	58–63	19–23	16–21
River ships (self-propelled)	$L \cong 80$ m	$L \cong 110$ m	78–79	69–75	11–13	13–19

W_L light ship weight, W_{ST} weight of steel structure, W_{OT} weight of outfitting, W_M weight of machinery installation

[a] Bulk carriers without own cargo handling equipment

[b] Crude oil tankers

[c] Triple E class of containerships of Maersk, DWT = 165,000 t, first launched 2013

[d] Banana reefers

[e] Oasis class cruise ship of Royal Caribbean Int., $L = 360$ m, 225,282 GT, launched 2009

the Ship Design Laboratory of NTUA (http://www.naval.ntua.gr/sdl). Illustrative examples of regressive analysis of basic characteristics for various types of ships are shown in Appendix A;

c. Using specific diagrams, for example (DWT/Δ) versus (DWT) and/or (speed) for various types of ships (see Figs. 2.1, 2.2, 2.3, and Appendix A).

It should be noted that for the volume carriers (Sect. 1.3.7.2), which are distinguished by their small DWT/Δ ratios, it is not appropriate to first estimate Δ with the above methods, nor at this initial stage, except for the cases for which there are robust comparative data from similar ships. In addition, further factors that also affect displacement, other than DWT, that is, type and required power of machinery system, the complexity of steel structure and the extent of outfitting, should be

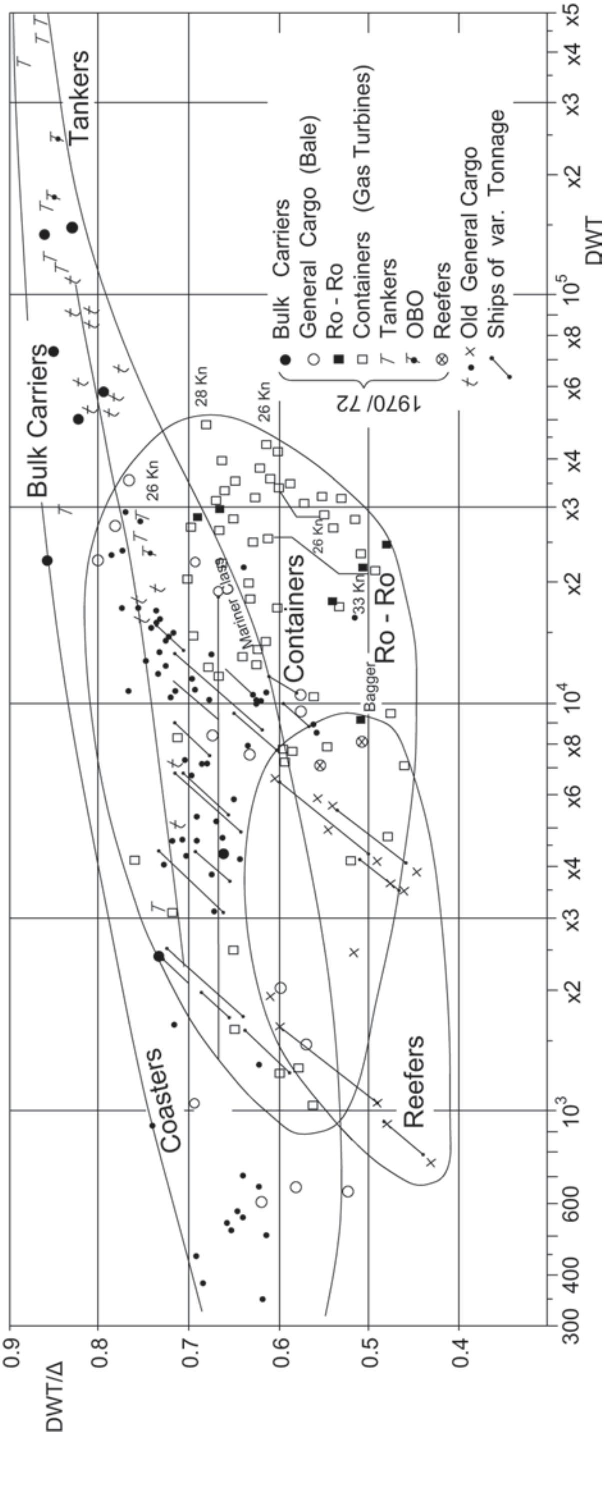


Fig. 2.1 (DWT/Δ) ratios versus DWT for cargo ships by Völker (1974)

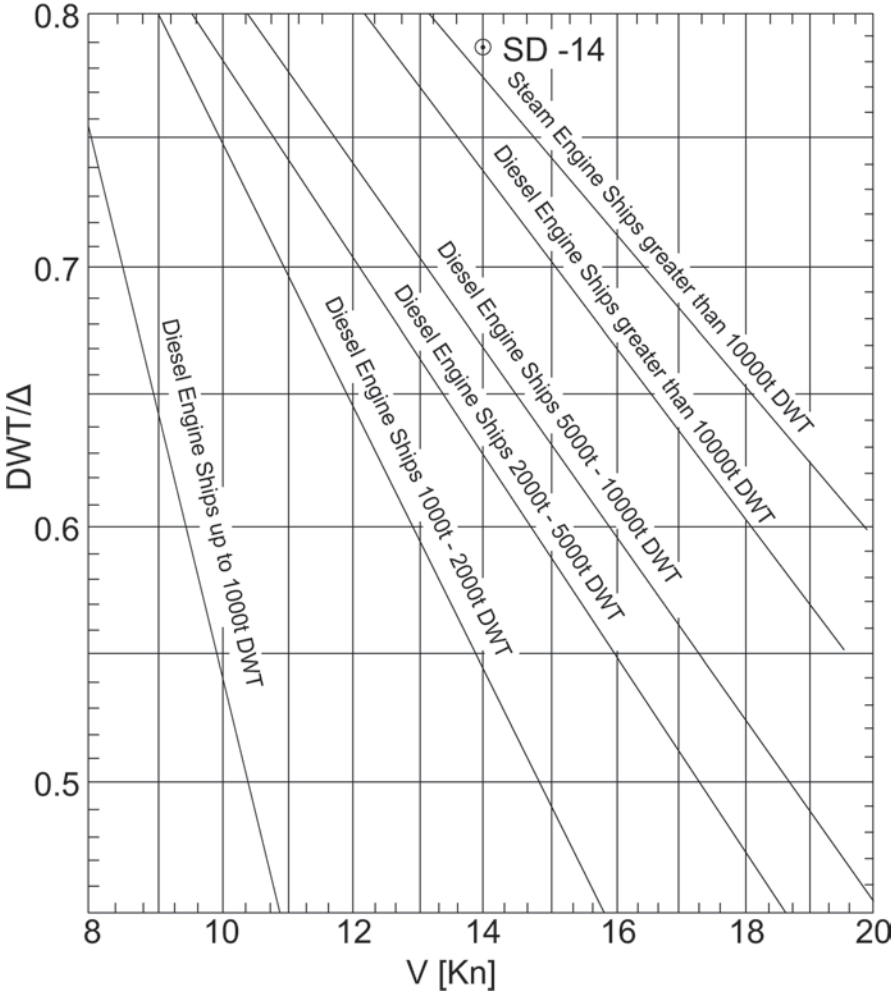


Fig. 2.2 Qualitative trend values of (DWT/Δ) ratios versus DWT and speed V for diesel engine ships by Schünemann (Henschke 1964)

checked with respect to possible deviation from typical/normal characteristics of comparative ships.

As described later on, it is possible to more accurately calculate the displacement by analysis of the various weight components that constitute the displacement weight Δ; however, this requires additional information from similar ships. E. Danckwardt's approximate method, though relying on past years' design practice, proved useful in related estimations of general cargo ships (see Papanikolaou 2009a).

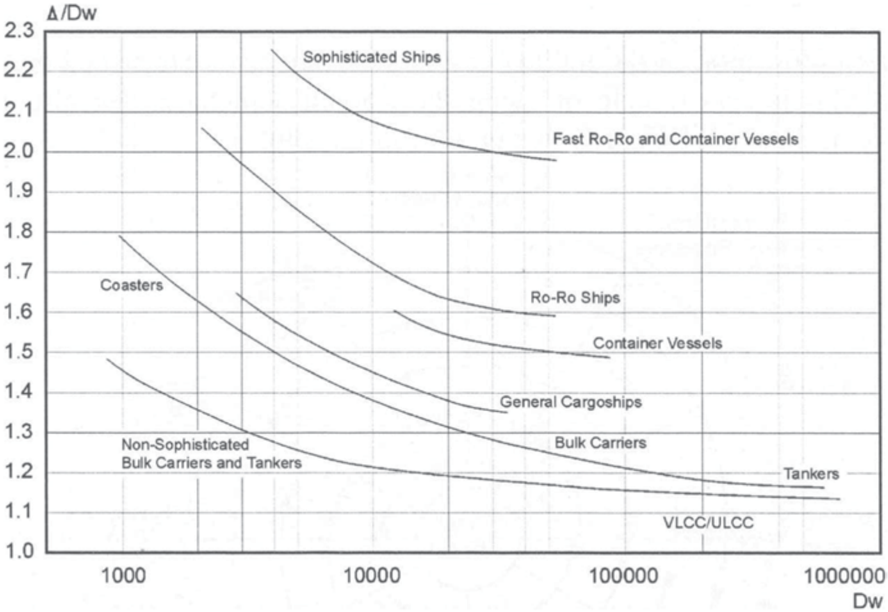


Fig. 2.3 (Δ/DWT) ratios versus DWT for various ship types, Harvald (1986) (see Friis et al. 2002)

2.2 Selection of the Main Dimensions and Form Coefficients

The procedure of determining the main dimensions, that is length L, beam B, draft T, side depth D, and hull form coefficients (initially the block coefficient $C_B$and then the other coefficients C_P, C_M and C_{WP}) should be conducted considering the following basic factors:

1. Ship's hydrodynamic performance (resistance and propulsion, seakeeping, maneuverability)
2. Satisfactory stability
3. Sufficient volume of cargo holds
4. Adequate structural strength
5. Construction cost

The common sequence of determining the main dimensions, form coefficients, and other basic sizes has been briefly described in Sect. 1.3.7. In this section we present first the general principles governing the selection of the main dimensions and secondly various useful semiempirical formulas, which are analyzed from both the phenomenological and scientific point of view; they express relationships of ship's main dimensions and ship's fundamental properties.

The main objective in the determination of the main dimensions is to fulfill the set shipowner's requirements, which mainly concern the following:

a. Transport capacity (DWT, payload, and cargo hold volume)
b. Service speed and endurance range
c. IMO and national safety regulations (SOLAS-IMO 2013b, MARPOL-IMO 2013a, ICLL 1988, etc.) and construction standards of a recognized classification society.

The fulfillment of the aforementioned requirements should be associated with the best possible economic (optimal) solution, in terms of the minimum cost for ship's construction and operation, or even with respect to more complex economic criteria, like required freight rate (RFR), net present value (NPV), and return on investment (ROI).

The selection of the main dimensions, that is, of length L, beam B, draft T, side depth D, and essentially of the freeboard F_b (=D−T), as well as of the block coefficient C_B, determines to which extent the under-design ship will satisfy the aforementioned owner's requirements. Typically, improper selections and combinations thereof for the basic dimensions are almost impossible to be corrected retrospectively; they generally lead to uneconomic and/or technically insufficient solutions.

The procedure of selecting the main dimensions and characteristic sizes is based on an iterative approach with appropriate sequence, for example, estimation of displacement, selection of length, determination of C_B, determination of the beam, draft and side depth. This order applies to deadweight carriers and should be adjusted accordingly for volume carriers (see Sects. 1.3.7.1 and 1.3.7.2).

The basic factors on determining the main sizes are summarized in the following:

1. **Length *L*:** This is a function of displacement and speed. It has a significant influence on the weight of steel structure and accommodation/outfitting, hence on the construction cost. Also, it strongly affects both the ship's calm water resistance and seakeeping performance (motions, accelerations, dynamic loads, added resistance, and speed loss in seaways).
2. **Block coefficient C_B:** This is a function of the Froude number and is influenced by the same factors as for the length L.
3. **Beam *B*, Draft *T*, side depth *D*:** The determination of these dimensions is actually coupled and is affected by the following basic factors:
   - hold volume (D)
   - stability (B)
   - required freeboard (D, T)
   - safety against flooding and capsize (B, D, T)
   - propulsive and manoeuvring devices (T)

The main dimensions L, B, and T are often affected as well by the topological *limits of the route*, that is, the dimensions of canals, ports, channels, and confined waters that the under-design ship needs to pass through. Mostly the restrictions are referring to allowable drafts.

Some typical dimensions of well-known canals and channels (maximum allowable ship dimensions) are:

Panama Canal	$L < 289.56$ m (in general for merchant ships)
	$L < 299.13$ m (passenger ships and containerships up to 5,000 TEU)
	$B < 32.31$ m (exceptionally 32.61 m, if $T < 11.28$ m)
	T < 12.04 m (as the maximum allowable draft for tropical fresh water TFW, as applicable)
Suez Canal	L: no limit
	$B < 71.02$ m (233 ft)
	$T < 10.67$ m (concerning stern draft in ballast condition)
	$T < 12.80$ m (maximum allowable draft for $B < 47.55$ m, concerning fully loaded voyages southbound)
	$T < 16.15$ m (maximum allowable draft for $B < 42.67$ m, concerning fully loaded voyages northbound)
Canal St. Lorenz (North America—Canada Great Lakes)	$L < 222$ m
	$B < 23$ m
	$T < 7.6$ m
Northeast Sea Channel (Nord-Ostseekanal—Northern Europe)	$L < 315$ m
	$B < 40$ m
	$T < 9.5$ m
Malacca Straits (between Malaysia Peninsular and Sumatra island)	$T < 25$ m

New Panamax maximum passing dimensions (expected, as of 2014): length: 366 m, width: 49 m, draft: 15.2 m, capacity of containers: 12,000 TEU

Finally, in rare cases, the ship length may be constrained by the length of slipways or docks of selected shipyards, with which the shipowner has long-term collaboration in new buildings and/or maintenance of his fleet.

For shaping the ship's hull form, both below the waterline and above, it is required to determine a series of other naval architectural characteristics that are either numerically identifiable sizes or typical qualitative features. It should be noted, however, that the shaping of the hull form cannot be reduced to the determination of certain individual characteristic numerals, but includes quantitative and qualitative interactions among them.

The main numerical values/quantities that describe the hull form of a ship (symbols and definitions according to ITTC (International Towing Tank Conference 2008) are:

a.1 The block coefficient, C_B
a.2 The midship section coefficient, C_M
a.3 The prismatic coefficient, C_P
a.4 The waterplane area coefficient, C_{WP}

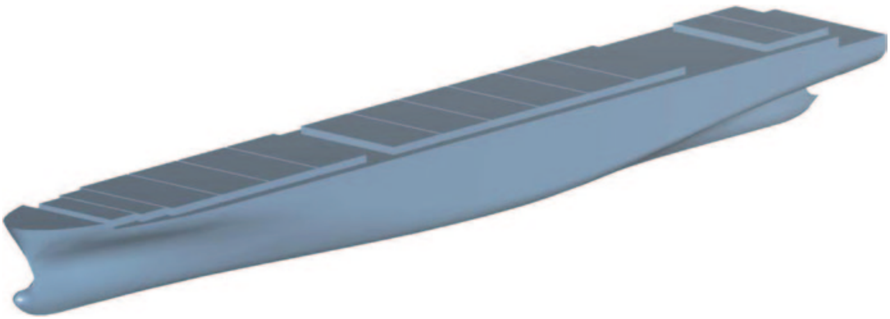


Fig. 2.4 Three-dimensional hull of a container ship designed with software TRIBON® at Ship Design Laboratory of NTUA

a.5 The slenderness ratio ($L/\nabla^{1/3}$) or the volumetric coefficient (∇/L^3)
a.6 The longitudinal center of buoyancy, $\overline{AB}$
a.7 The vertical position of center of buoyancy above baseline, $\overline{KB}$
a.8 The parallel body length, L_P
a.9 The length of entrance/run of sectional areas, L_E/L_R

The above sizes will be discussed in subsequent paragraphs.

The qualitative characteristics, which supplement the determination of the hull form of a ship, are:

b.1 Sections' character below waterline
b.2 Sections' character above waterline
b.3 Shaping of bow section (bow type, profiles of waterlines and sections in bow region, bulbous bow)
b.4 Shaping of stern section (stern type, profile of waterlines and sections in stern region, stern bulb, flow to propeller and rudder)
b.5 Freeboard and sheer deck

These features will also be discussed in subsequent paragraphs (Fig. 2.4).

2.3 Selection of Length

Satisfaction of the owner's main requirements (with respect to transportation capacity, service speed, endurance/range, and safety regulations) is possible with different choices of ship length. However, it is logical to look ultimately for the optimal length with respect to some economic criteria determined by the interests of the yard and/or the owner. In the first case, the employed economic criterion is the "minimum construction/building cost", whereas in the second case, ship's economy is generally evaluated by the "minimum required freight rate (RFR) per ton of cargo" criterion.

Two examples of optimization of the ship length with respect to the "minimum construction cost" and alternatively the "maximum return on investment" are given

in Papanikolaou (2009a, Vol. 2). From the available data, it is concluded that for fixed/given hold volume and displacement, increasing the length generally leads to an increase of the ship's structural weight and to a reduction of the ship's required propulsion power for achieving the specified speed.

As to the effect of a length increase on the other ship weight components (for fixed displacement), it also increases the accommodation/outfitting weight, what generally leads to a reduction of the ship's payload. The resulting reduction of propulsion power and the corresponding reduction of machinery and fuel weights, cannot balance the increases of the other weight components; thus, in order to maintain a certain payload level specified by the shipowner, it is required to increase the displacement, what induces some increase in propulsion power (proportional to $\Delta^{2/3}$), etc.

Regarding the building cost, the increase of length implies an increase of the steel cost, while a limited reduction of the cost of machinery propulsion system may be expected (see Chap. 6: estimation of shipbuilding cost). In simple approaches (apart from parametric mathematical optimizations) , the identification of the optimum, most economical solution may be accomplished by systematic variation of the ship's length around an estimated initial length. The latter results from comparisons with similar ships, by use of empirical diagrams or semiempirical formulas (see Appendix A and examples in Papanikolaou (2009a, Vol. 2).

2.3.1 *Effect of Length on Resistance*

It is assumed that, the total resistance R_T of a ship, with a wetted area S, sailing at speed V in calm water of density ρ, can be decomposed according to the hypothesis of W. Froude[1] (1868) as follows:

$$R_T = R_F + R_R \tag{2.1}$$

where R_T is the **T**otal **R**esistance or **T**owing **R**esistance, which has two components,

- the Frictional Resistance R_F and
- the Residuary Resistance R_R

that are elaborated in the following.

The qualitative characteristics of the per ton displacement total ship resistance and of its main components for various speed–length ratios $V\ (kn)/\sqrt{L\ (\mathrm{ft})}$ are illustrated in the following graph (Fig. 2.5).

The frictional resistance is determined as

$$\text{Frictional resistance: } R_F = \tfrac{1}{2} C_F \rho S V^2 \tag{2.2}$$

[1] William Froude (1810–1878) Eminent English engineer, naval architect and hydrodynamicist; he was the first to formulate correctly the law for ship's water resistance and to set the foundations for modern ship model testing, by introducing a unique dimensionless similitude number (Froude number) by which the results of small-scale tests could be used to predict the behaviour of full-sized ships; of importance are also his contributions to ship's stability in waves.

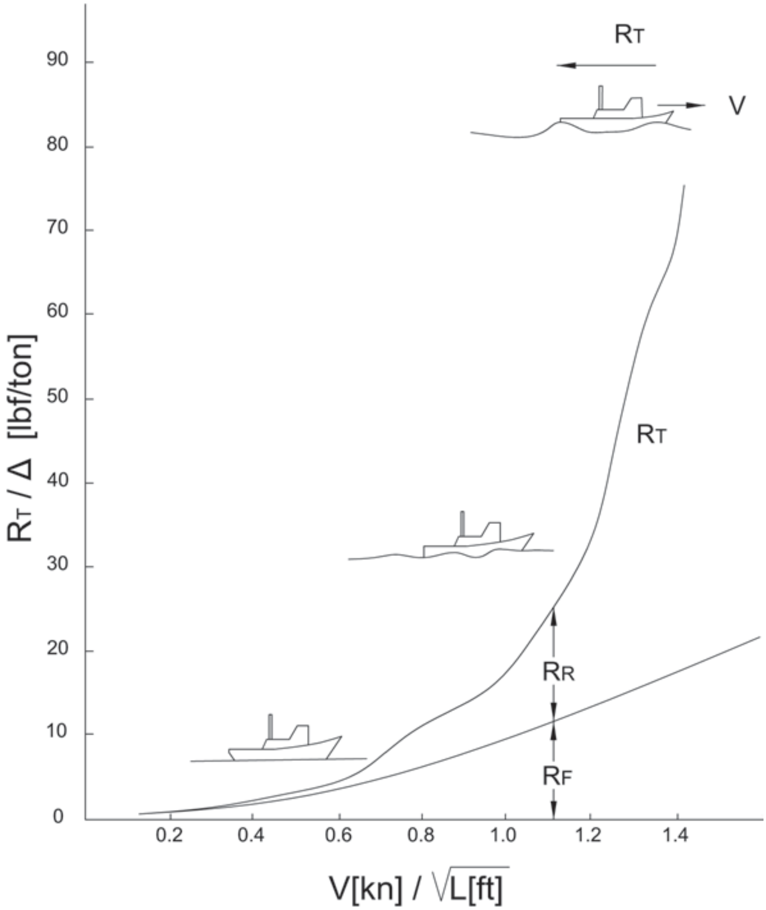


Fig. 2.5 Typical total resistance (per ton displacement) curve as a function of the speed–length ratio $V/\sqrt{L}$ for displacement ships (without dynamic lift)

where

$C_F = f(R_n)$: nondimensional frictional resistance coefficient dependent on the nondimensional Reynolds number, that is, $R_n = V \cdot L/\nu$, ν: sea water's kinematic viscosity ($=1.19 \cdot 10^{-6}$(m²/s) at 15 °C), $L = L_{WL}$, V ship's speed (m/s).

$C_F = 0.075/(\log_{10} R_n - 2)^2$
according to ITTC 1957.

S: wetted hull surface, $\approx (3.4 \cdot \nabla^{1/3} + 0.5 L_{WL}) \cdot \nabla^{1/3}$ according to Lap (Figs. 2.6 and 2.7).

$$\text{Residuary resistance } R_R = \tfrac{1}{2} C_R \rho S V^2 \tag{2.3}$$

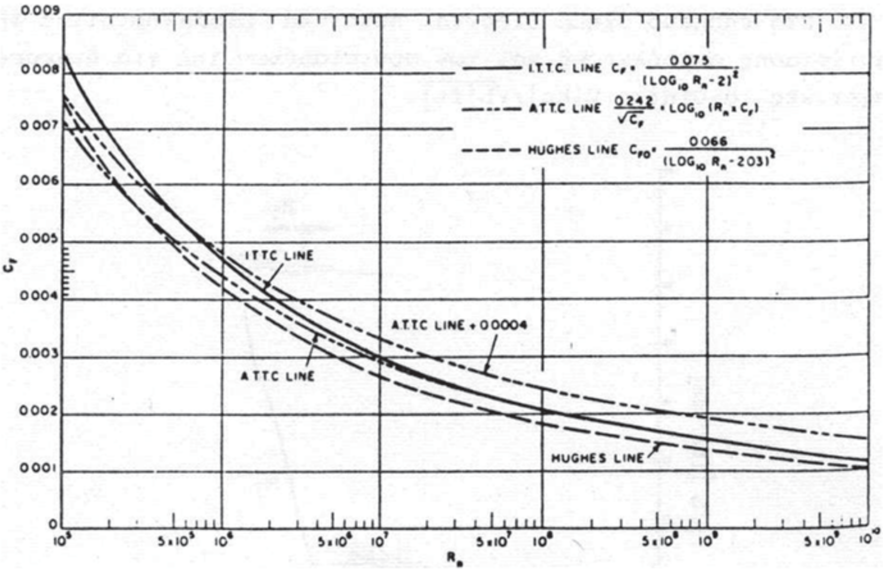


Fig. 2.6 Basic relationships for calculating the ship's frictional resistance coefficient, $C_F = f(R_n)$. (Lewis 1988)

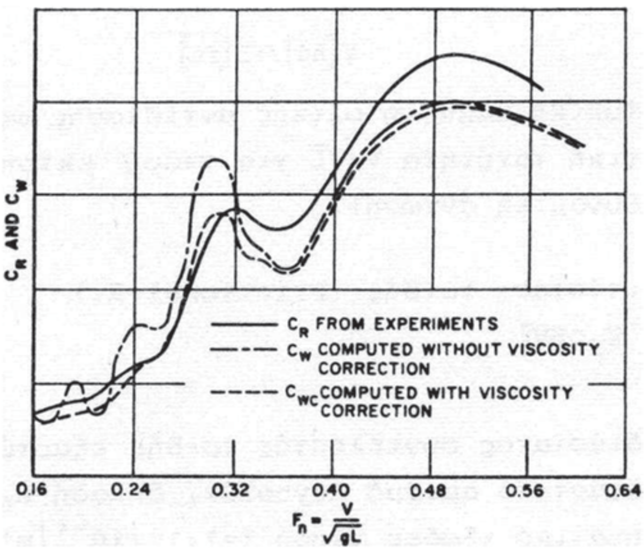


Fig. 2.7 Qualitative relationships of the residuary resistance coefficient C_R and wave resistance coefficient C_W with Froude number F_n—Comparison of results from model experiments and numerical estimations. (Lewis 1988)

where

$C_R = f(F_n, R_n)$: nondimensional residuary resistance coefficient, which is dependent on Froude number $F_n = V\sqrt{g \cdot L}$ (where g is gravitational acceleration), on Reynolds number, *and* on the ship's hull *form* (*Form Resistance*)

The residuary resistance R_R can be further decomposed as follows:

$$R_R = R_W + R_{PV} \tag{2.4}$$

where R_W is the wave resistance,

$$R_W = \tfrac{1}{2} C_W \rho S V^2 \tag{2.5}$$

C_W is the nondimensional wave resistance coefficient,

$= f(F_n$, hull form)

R_{PV} is the pressure viscous[2] resistance,

$$R_{PV} = \tfrac{1}{2} C_{PV} \rho S V^2 \tag{2.6}$$

C_{PV} is the nondimensional pressure viscous resistance coefficient,

$= f(F_n, R_n$, hull form)

As mentioned earlier, the residuary resistance R_R and the corresponding coefficient C_R are functions of both the F_n and R_n numbers, and of the ship's hull form.

In Froude's original, simplified hypothesis, it was assumed that

$$C_R = f_1(F_n) + f_2(F_n, R_n) \cong f_3(F_n) \tag{2.7}$$

that is, the effect of R_n on the residuary resistance is neglected.

If we consider the wetted surface area S approximated according to the simplified Taylor's formula (Lewis 1988)

$$S = C_S \cdot \sqrt{\nabla \cdot L} \tag{2.8}$$

where $C_S = f(B/T, C_M)$, and the frictional coefficient C_F is taken for turbulent flow according to Prandtl:

$$C_F = 0.072 \cdot R_n^{-0,2} \tag{2.9}$$

[2] Sometimes called "form" resistance, though correctly the *residuary* resistance is ship's "*total form dependent* resistance or *pressure* resistance."

then it is concluded that, for an increase of length with the ratio:

$$\lambda = L_1 / L_0 \tag{2.10}$$

where $(...)_0$ holds for the parent hull and $(...)_1$ for the present hull, the frictional resistance R_F increases with the ratio:

$$(R_F)_1 / (R_F)_0 = \lambda^{3/10} \tag{2.11}$$

Assuming the residuary resistance coefficient to be a function of F_n number:

$$C_R = f(C_n) \cong C \cdot F_n^{\alpha}, \tag{2.12}$$

where the exponent α is typically taken between 3 and 5, depending on F_n and hull form, the ratio for the residuary resistance is concluded:

$$(R_R)_1 / (R_R)_0 = \lambda^{-(\alpha-1)/2} \tag{2.13}$$

where $3 \leq \alpha \leq 5$

thus, a reduction of the residuary resistance by the ratio λ^{-1} to λ^{-2}.

For the total resistance it follows:

$$(R_T)_1 = (R_F)_0 \cdot \lambda^{3/10} + (R_R)_0 \cdot \lambda^{-(\alpha-1)/2} \tag{2.14}$$

Therefore, for typical ship lengths L and Froude numbers $F_n \geq 0.15$ the *reduction* of the *residuary resistance* R_R *with an increase* of λ is more drastic *than the increase of the wetted surface*, and of the frictional resistance R_F, resulting in *the decrease of the total resistance*.

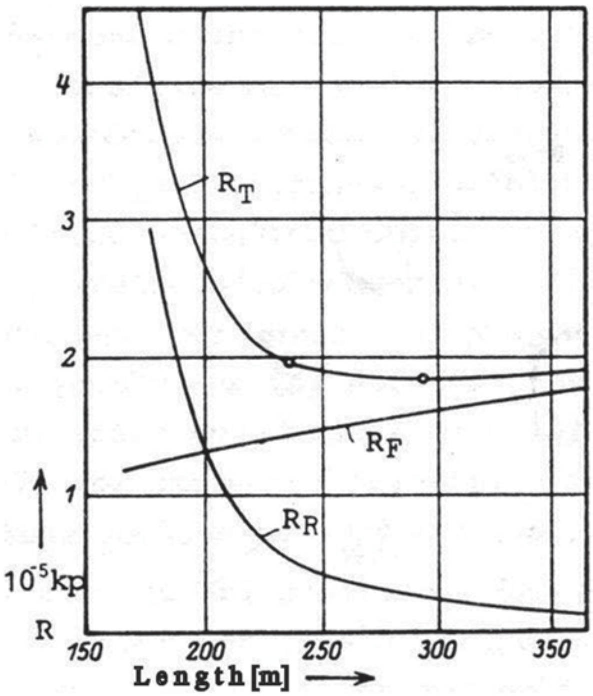
The historical Fig. 2.8 from David W. Taylor (1943), which is based on the analysis of systematic towing experiments of ship models for full scale naval vessels of *constant* displacement 30,000 t and 29 kn speed, shows the minimum total resistance for a length of L ~ 300 m, as well as the drastic reduction of the residuary resistance with the increase of length.

Obviously, the trend of these curves may change for other ships, according to the percentage share of the R_R and R_F components in the total resistance R_T (Fig. 2.8).

Thus, for small Froude numbers (≤ 0.15), as is the case for example for tankers/bulk carriers, the frictional resistance constitutes the primary part of the total resistance (~ 80 % R_T), while for relatively fast ships ($F_n > 0.25$), the conditions are just reversed (see following figure and Table 2.3) (Fig. 2.9).

Apart from the indirect influence of length on the R_R and R_F resistance components, it is important to attempt to avoid unfavorable Froude numbers, around which the superposition of the primary bow and stern wave systems leads to tuning, resulting in an increased wave resistance R_W. This means, when the wave crest/trough of the generated/shipbound bow system coincides with the corresponding crest/trough of the stern system, this leads through superposition to a tuning, re-

Fig. 2.8 Effect of length on the resistance of a ship with constant displacement Δ=30,000 t and speed *V*=29 kn according to DW Taylor (1943)



sulting in a wave system of increased wave amplitude; consequently, it causes an increase of wave resistance. The latter corresponds to the energy loss of the ship in view of the disturbance of the calm water surface and it is proportional to the square of the amplitude of the generated waves. These phenomena are now elaborated in more details.

A: Slow cargo ship
B: Transoceanic passenger ship
C: Small passenger ship or cruiser
D: Torpedo boat

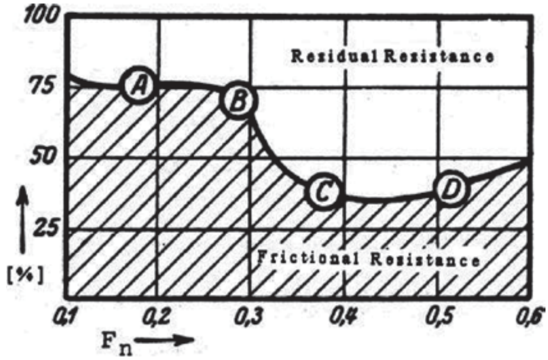


Fig. 2.9 Percentage shares of frictional resistance and residuary resistance for different ship types and characteristic Froude numbers by F. Horn (1930). *A* Slow cargo ship, *B* Transoceanic passenger ship, *C* Small passenger ship or cruiser, *D* Torpedo boat

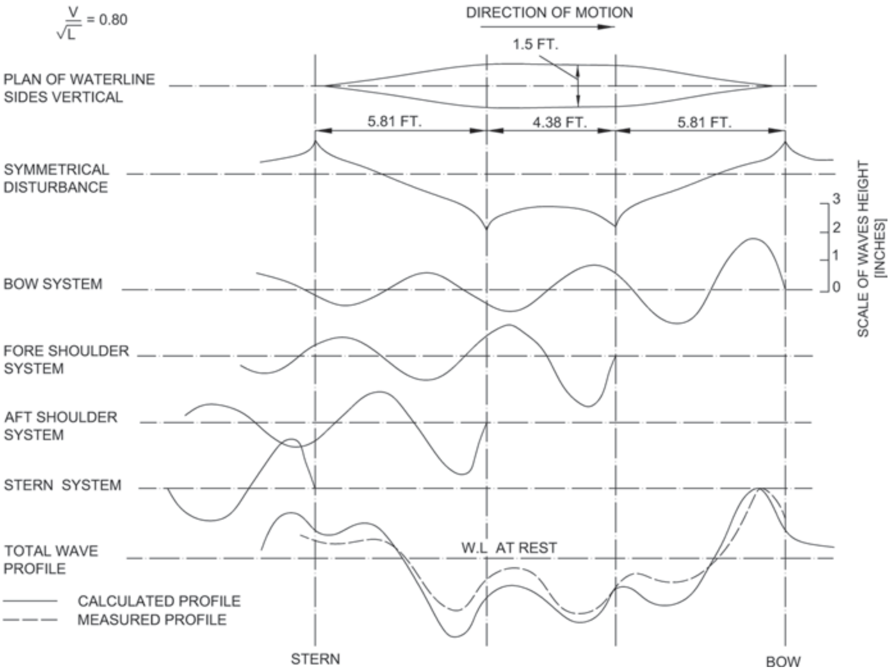


Fig. 2.10 Wave systems for a double-wedged model according to Wigley. (Lewis 1988)

It is well-known from analytical and experimental studies that the symmetrical pressure distribution arising around a double-wedged body, with parallel midbody (see Fig. 2.9 according to Wigley in Lewis 1988), which sails with constant forward speed on the calm water surface, is the cause for initially two wave crests at the bow and aft perpendiculars and an extended trough along the ship's parallel midbody (Fig. 2.10a).

The system (a) as shown in Fig. 2.10, which is also known as "primary wave system," travels at the same speed as the vessel, so that it stays at the same position with reference to the moving ship; due to the double symmetric pressure distribution around the ship, this wave system does not cause any resistance as long as the ship moves with constant forward speed (assuming inviscid, ideal flow). However, this primary wave system is the underlying cause for the following four "secondary" wave systems:

1. The bow wave system (b), starting with a crest
2. The fore shoulder system (c), starting with a trough
3. The aft shoulder system (d), starting with a trough
4. The stern wave system (e), starting with a crest

Considerably behind the ship, all four above secondary systems (b) to (e) acquire pure sinusoidal form of decaying amplitude and a length that corresponds to the

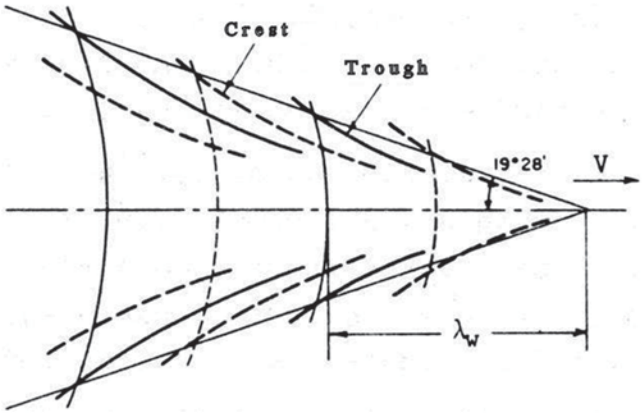


Fig. 2.11 Wave systems behind a pressure point moving at constant speed V, according to Lord Kelvin (1887). (see Lewis 1988)

length of a free surface wave travelling at the same speed V as the ship. For such waves on the free water surface the following relationship applies:

$$\lambda_w = (2\pi / g)V^2 \tag{2.15}$$

where λ_w = length of a free surface wave with velocity V in deep water.

According to the theory of Lord Kelvin (1887) a singular "pressure point" moving on a straight line with velocity V on the free water surface creates two wave systems behind itself (Fig. 2.11). As seen in the figure, the characteristic model of the Kelvin waves is composed of two subsystems:

- One *transverse* system that starts with a crest or trough at the pressure point (depending on the pressure value, positive or negative) and which has a wavelength, as given above; crests and troughs are indicated by dotted and full lines; and
- One *diagonally moving divergent* wave system bounded by two straight lines forming an angle of 19°28′ (deep water hypothesis) with respect to the straight travelling line; crests and troughs are indicated as above.

Assuming the aforementioned "primary" pressure system (a) in Fig. 2.10 composed of an infinite number of Kelvin "pressure points", then obviously the number of the generated secondary systems (b) to (e) increases accordingly. However, even the simplified modeling/superposition of only two basic waves, namely that of the bow wave (b) and that of the stern wave (e), leads to the essence of the reasoning regarding the causes of wave resistance (Figs. 2.12 and 2.13).

As shown in the detailed Fig. 2.10, the superimposition of the secondary wave systems leads to a non-symmetric profile of the wave surface (and of the pressure distribution) around the ship. Due to the corresponding non-symmetric pressure distribution, a net longitudinal force results, opposite to the direction of the ship's motion. This force is known as "wave (making) resistance".

If the ship speed changes, the length of the secondary wave systems will change accordingly, whereas the wave generation points remain unchanged; this leads to

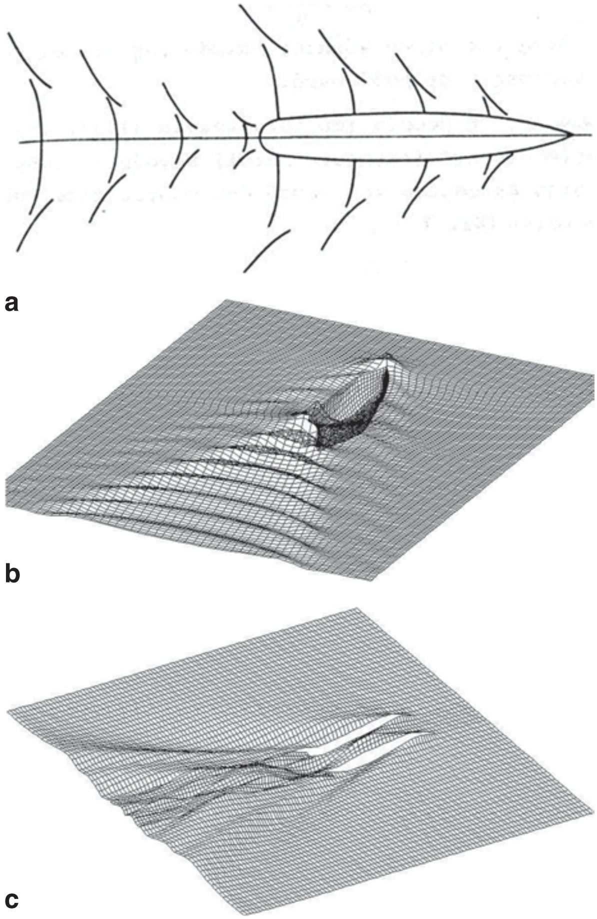


Fig. 2.12 **a** Schematic diagram of simplified superposition of the ship's bow and stern wave systems. **b** Wave systems of a container ship, generated numerically with 3D panel method (SHIPFLOW). **c** Wave systems of a catamaran, generated numerically with 3D panel method (SHIPFLOW®)

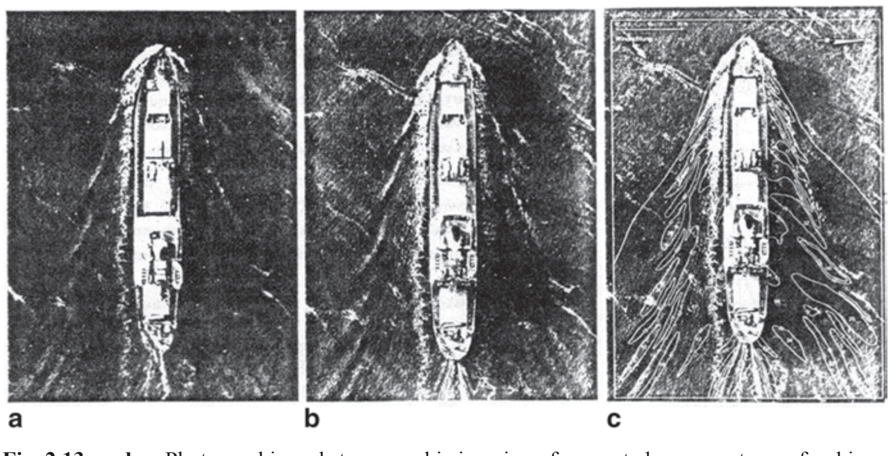


Fig. 2.13 **a, b, c** Photographic and stereographic imaging of generated wave systems of a ship

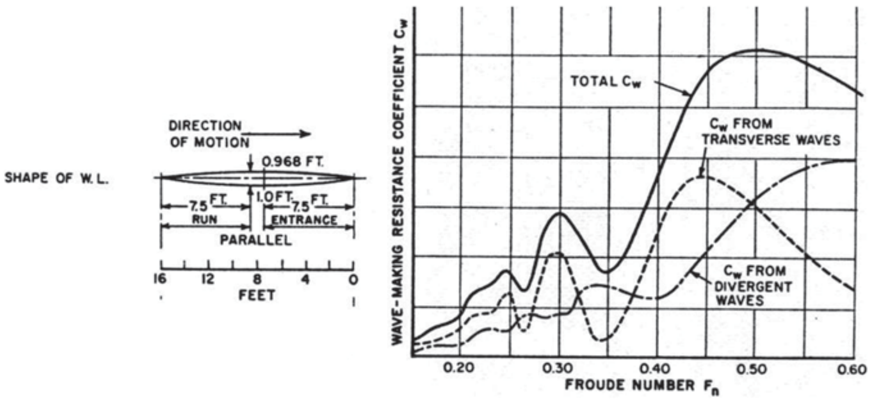


Fig. 2.14 Analysis of the components of wave resistance for a ship model with parabolic waterline. (Lewis 1988)

a modification of the resulting superposition of the wave systems and of the corresponding wave resistance.

The nondimensional wave resistance coefficient takes the form:

$$C_{\mathrm{w}} = \frac{R_{\mathrm{w}}}{\frac{1}{2} \cdot \rho V^2 \cdot S} \tag{2.16}$$

Apart from its dependence on the ship's hull form, it shows strong fluctuations as a function of the ship's speed and of Froude number, in accordance with the outcome of the superposition of the secondary wave systems (tuning or attenuation). A typical example for the behavior of $C_{\mathrm{w}}=f(F_n)$ for a ship model with parabolic waterline is given in Fig. 2.14; it shows the contributions from the transverse and divergent wave systems to total wave resistance.

In order to obtain a *favorable* ship operational region with respect to Froude number, that is, to have a relatively reduced wave resistance, we need to ensure at least the tuning of the bow (b) and stern (e) wave systems so that they cancel each other, thus to achieve wave attenuation. Mathematically, the ratio of waterline length L_{WL}, which corresponds approximately to the distance of the wave generation points of the two systems, to the half wavelength λ_w must be an ***odd*** number, namely:

$$L_{\mathrm{WL}} / (0.5\lambda_w) = (2n+1), \quad n = 1,\ 2,\ 3\ \ldots \tag{2.17}$$

On the contrary, for *adverse* Froude number operational regions this ratio should be an ***even*** number.

We present in Table 2.2 the adverse and favorable regions of Froude numbers, as derived from model experiments; they are approximately confirmed by applying the above simplified relationships between L_{WL} and λ_{w}.

If during the selection of the ship's length it is found that the ship's operating region is located within the limits of unfavorable Froude numbers, it is possible to

Table 2.2 Adverse and favorable regions of Froude numbers in terms of wave resistance for normal ships

	$F_n = V / \sqrt{g \cdot L_{PP}}$
Adverse regions (tuning)	0.45–0.50, 0.29–0.31, 0.23
Favorable regions (attenuation)	0.33–0.36, 0.25, 0.21

avoid or mitigate the undesirable interference of the generated wave systems with the following measures:

1. Change of length[3]
2. Smoothing of hull shoulders
3. Change of speed

Table 2.3 presents typical operating points of common ship types. It can be observed that the operating points of certain vessels are in the undesirable Froude number regions. However, it is noted that in these cases either the percentage share of R_W in total resistance is small (low Froude number), or it concerns ships of medium to small absolute speeds combined with small lengths (fishing vessels). As regards naval ships, with $F_n > 0.5$, there it is attempted to mitigate the tuning phenomena of the bow and stern wave systems with appropriate smoothing of the hull.

Critical-boundary and economic speed We may select the appropriate length in conjunction with the hull form coefficients C_B or C_P and the corresponding speed, using the basic formula of Alexander (see Eq. 2.1):

$$V(\text{kn}) / \sqrt{L_{PP\,(\text{ft})}} = 2(K_1 - C_B) \tag{2.18}$$

Table 2.3 Percentage contribution of frictional resistance to the total resistance and typical operating points of modern ships in terms of Froude number. (Schneekluth 1985)

F_n	(R_F/R_T) (%)	$C_W = f(F_n)$	L^*/λ_W	$L^*/(0.5\lambda_W)$	Ship type
0.15	80	Crest	5.0	10	Large size tanker (VLCC)
0.19	70	Trough	4.5	9	Medium size tanker
0.23	60	Crest	3	6	Medium speed cargo ship
0.25	60	Trough	2.5	5	Fast cargo ship
0.29–0.31	50	Crest	2	4	Fishing ship
0.33–0.36	40	Trough	1.5	3	Fast cargo ship/Reefer
0.40		Crest	1	2	
0.50	30÷35	Crest	0.64	1.28	Naval ship/cruiser
0.563		Trough	0.5	1	

L^* distance of the crest of bow wave to trough of stern wave system, $L^* \cong L_{WL}$
λ_W length of generated waves $= (2\pi/g)V^2$

[3] However, an increase of length has typically significant negative side effects on some ship weight components (especially on structural weight and payload) and construction cost, as has been stated already.

where

$K_1 = 1.08$ for trial speed V_T (trial)
$= 1.05$ for service speed V_S (service),

or Troost's formula:

$$V_S(\text{kn}) / \sqrt{L_{PP\,(\text{ft})}} = 1.85 - 1.6C_P \tag{2.19}$$

where

V_S: service speed,
$\cong 0.94\ V_T$ (trial speed),

In this respect, we define a speed limit (boundary or critical velocity) in relation to the ship's characteristics, expressed here by L and C_B, the exceedance of which leads to a rapid increase of the required propulsion power.

It can be shown that while the part of the required propulsion power, which corresponds to the frictional resistance, increases approximately with the exponent 2.8 with respect to speed, the corresponding residuary resistance has an exponent that may be even more than 5. Thus, it is concluded that for the propulsive power P:

$$\mathrm{P} \propto V^n, \text{where } n \geq 3$$

As a *boundary* or *critical* speed we define the speed the excess of which is related to an exponent greater than 3:

$$n(V \leq V_{CR}) \cong 3$$

A simple descriptive explanation for the *boundary speed* is the abrupt drop of the British Admiralty constant at that speed:

$$C_N = \frac{V^3 \Delta^{2/3}}{P} \tag{2.20}$$

see $C_N = f(V)$, Fig. 2.15 (F. Horn[4] 1930). Likewise, we may see the same effect by observation of rapid increase of the *circular* total resistance coefficient ©, namely

$$© \equiv \frac{R_T}{\Delta} \cdot \frac{1000}{\mathrm{K}^2} \quad K \equiv 0.5834 \frac{V}{\Delta^{1/6}}$$

$©_{400FT}$: refers to ships with a standard length of 400 ft.

R_T (tonf), Δ (tonf), V (kn)

Anglo-Saxon units: 1 tonf = 1 long ton = 1.016 metric tons.

[4] Fritz Horn (1880–1972): Eminent German professor of ship theory at the Technical University of Berlin; before becoming professor in 1928, he worked for the German shipbuilding industry and the navy; his main contributions are in the theory of antirolling tanks, propeller theory, including the theory of ducted propellers, and wave induced vibrations.

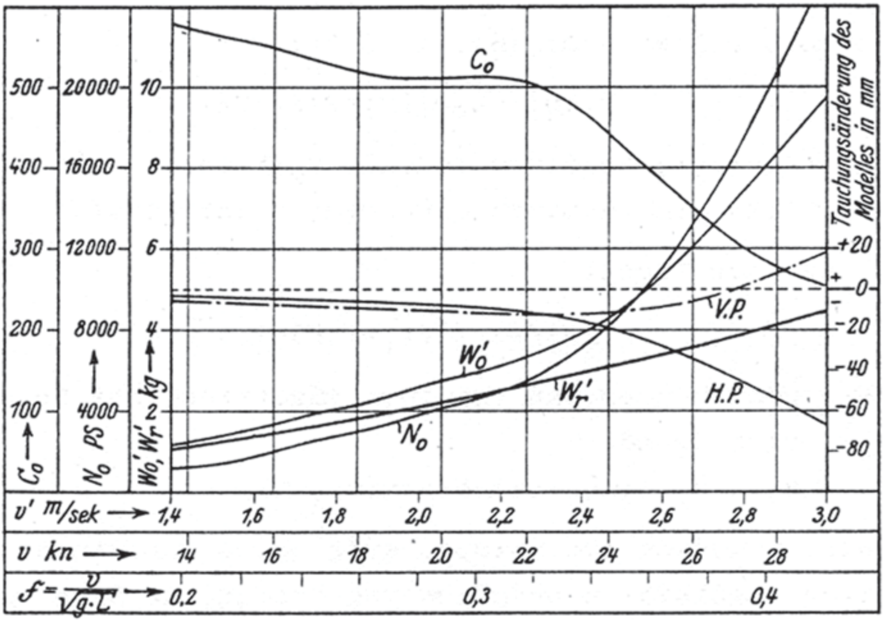


Fig. 2.15 Results of towing experiments of ship model according to Horn (1930). ($C_0 \equiv C_N$ (Admiralty constant), $N'_0 \equiv P_E$ (effective power), $W'_0 \equiv R_T$, $W'_r \equiv R_F$, ( )′ means: model values, V.P.: draft at forward perpendicular, H.P.: draft at aft perpendicular

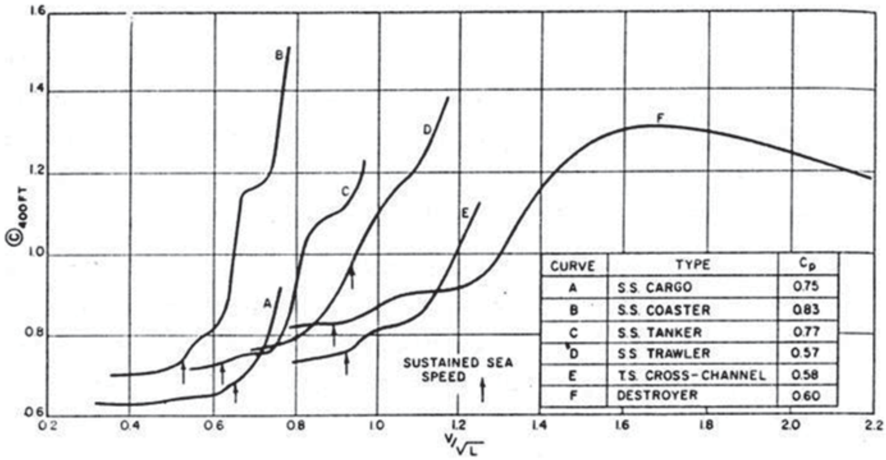


Fig. 2.16 Characteristic curves of the Anglo-Saxon *circular* total resistance coefficient © for various types of ships with indication of service speed. (Lewis 1988)

It is logical to note that the "service" speed is (almost) always chosen to be smaller than the "critical" speed by a certain percentage, depending on the form of the curve $P=f(V)$ or $C_w=f(F_n)$ (see Fig. 2.16). In this way the ship can be operated economically in speed regions with relatively reduced resistance, and it is possible to

recover potential delays during a voyage, that is, by slight increase of the speed without significant increase of the required propulsion power (and fuel consumption).

The trial speed of a ship, which is regarded approximately 6% higher than the service speed, is usually very close to the critical speed. Therefore, *assuming that the operational conditions are identical to those at the trials* (in particular: calm and deep water; clean hull; no wind, waves, or currents), it is concluded:

$$P(V_{\mathrm{T}}) \cong 1.25 P(V_{\mathrm{S}}) \tag{2.21}$$

as long as the trial speed $V_{\mathrm{T}} \cong 1.06\ V_{\mathrm{S}}$, and the corresponding horsepower increases with an exponent of at least 4 in terms of speed.

However, under service conditions there is normally a resistance increase caused by hull fouling, weather conditions, etc., which is in the range of 10–25%. Thus, it may be argued that, in practice *it requires approximately the same horsepower* as for the 6% higher trial speed to achieve the *service speed under service conditions*.

Conclusions

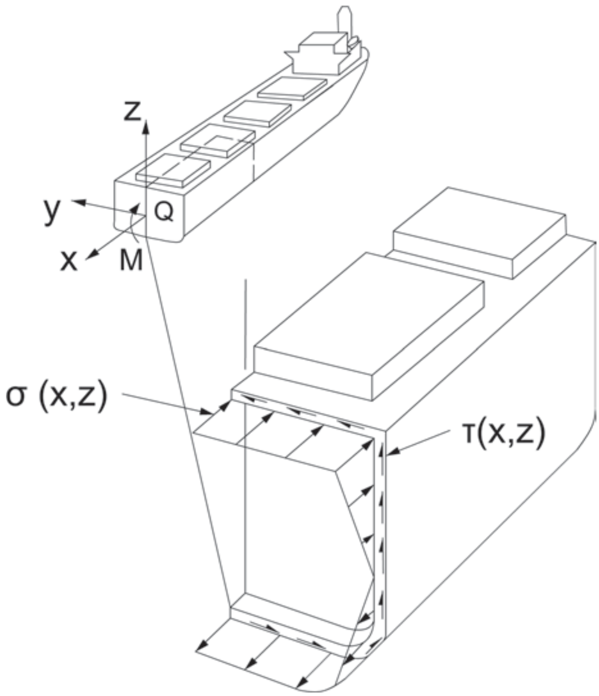
1. The selection of the ship's length, with the least resistance/powering criterion in mind, is based on the *value of the ship's relative speed*, that is, *the Froude number* that correlates the speed with the length, rather than on the *absolute* speed.
2. Relatively slow vessels, with a small operational Froude number (up to about 0.20), exhibit a high percentage of frictional resistance, in relation to their total resistance (see Table 2.3) and require, for the reduction of frictional resistance, minimum wetted surface for given displacement, which geometrically corresponds to short and full hull forms[5], that is, very high C_{B} and C_{P} coefficients, but relatively small lengths L and low slenderness coefficients $L/\nabla^{1/3}$.
3. Relatively fast ships ($F_n \geq 0.25$), on the contrary, with a significant proportion of wave resistance, require relatively slender hulls, that is, low C_{P} and C_{B} coefficients, high slenderness coefficient $L/\nabla^{1/3}$, appropriate lengthwise distribution of displacement, with the center of buoyancy abaft of midship and relatively large lengths L.

2.3.2 *Effect of Length on the Ship's Strength and Structural Weight*

In order to explore the influence of length on the ship's longitudinal strength and structural weight, we consider the ship in a simplified manner, namely as *a slender*

[5] Note that the hull form with smallest wetted surface for given displacement volume is the sphere (or floating half-sphere). This fact led recently designers to look into innovative hull forms of slow steaming cargo ships with ellipsoidal characteristics around amidships, which proves beneficial both with respect to low resistance and minimum ballast water requirements (see the E4 containership concept by G. Koutroukis and A. Pavlou of NTUA-SDL, VISIONS European academic competition 2011).

Fig. 2.17 Consideration of the ship's hull as a bending beam



bending beam (see Fig. 2.17), for which the following relationship applies (Fig. 2.18):

$$\sigma(x,z) = M(x) \cdot z \,/\, I(x) \tag{2.22}$$

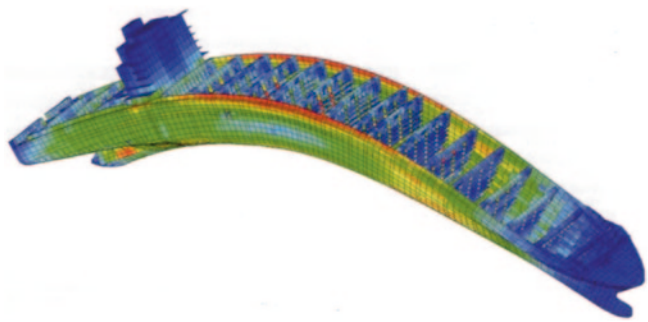


Fig. 2.18 Bending deflection (vertically magnified) of a containership in hogging state—calculated stress distribution (*colored levels*) by Finite Element Method (FEM). (Source: Germanischer Lloyd)

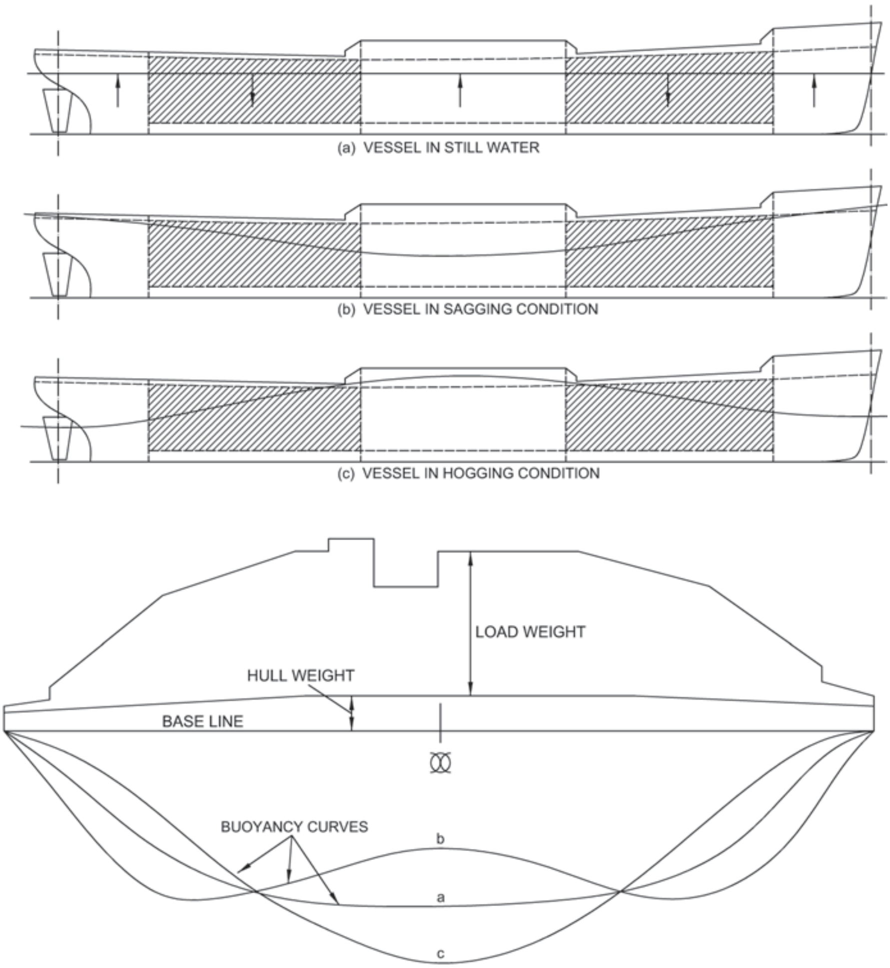


Fig. 2.19 Typical loading conditions and associated buoyancy and load distributions for the longitudinal strength of a ship (Lewis 1988)

where

σ: bending stress at point (x, z)
M: bending moment at point (section) x
I: moment of inertia of the beam's (ship's) cross-section at x

The slender bending beam assumption for the ship is acceptable particularly for ships with high L/B and L/D ratios. The high levels of bending stress arise at the midship region in case of both *still water bending* (Fig. 2.19a) and for typical *bending of the ship in waves*.

In particular, two extreme situations are usually examined, namely considering the ship ("frozen") as travelling on the crest or the trough of a following/stern wave of approximately the same length and speed as the ship; in the first case a buoyancy

excess presents around the midship section, when the crest of the loading wave is close to the midship section (*hogging*); in the second case, the buoyancy excess is situated at both ends of the ship, which implies that the wave trough is at the centre of the ship (*sagging*) (Fig. 2.19b and c).

The maximum bending moment occurs at the midship section and it can be approximated by the following approximate formula (see Strohbusch[6] 1971):

$$M_{\max} = M(x = L/2) = C\Delta L \tag{2.23}$$

where

C: constant dependent on ship type, loading condition, wave length and height. Typical approximate values for the full load condition and main engine amidships or slightly abaft are listed below.

For fully loaded cargo ships in general we have (approximately):

C $=0.012$ (calm water)—generally for cargo ships
$=0.025$ ("hogging", additional moment)
$=-0.013$ ("sagging", additional moment).

For fully loaded tankers we have:

C $=-0.006$ to $+0.003$ (calm water)
$=0.020$ ("hogging", additional moment)
$=-0.028$ to -0.020 ("sagging", additional moment).

In the process of the ship's preliminary design, a more precise examination of the ship's longitudinal strength is required, namely the evaluation of the sum of acting bending moments (and of vertical shear forces) under the various loading conditions and for navigating in both calm water and waves, according to the specifications of recognized classification societies.

Among these conditions, the maximum still water bending moment and vertical shear forces result from the differences between the longitudinal distributions of buoyancy and weight of the ship on the basis of refined hydrostatic calculations, which are routinely conducted by use of standard naval architectural software packages.

For the additional bending moments at midship from the loadings in waves the latest specifications of the International Association of Classification Societies IACS (IACS UR S-11) can be used, namely

$$M_{\mathrm{WS}}\ (\mathrm{kNm}) = -110\ C\ L^2\ B(C_{\mathrm{B}} + 0.7)10^{-3} \quad \text{(sagging moment)} \tag{2.24}$$

$$M_{\mathrm{Wh}}\ (\mathrm{kNm}) = +190\ C\ L^2\ B\ C_{\mathrm{B}}10^{-3} \quad \text{(hogging moment)} \tag{2.25}$$

[6] Erwin Strohbusch (1904–1980) Leading German professor of ship design at the Technical University of Berlin after WWII; before becoming academician, he worked as naval architect in leading positions at the German Navy and at the Henschel aircraft industry as aerodynamicist and aircraft designer.

and

$$C = 10.75 - (3 - L/100)^{1.5} \qquad \text{for } 90 \le L \le 300 \text{ m}$$
$$= 10.75 \qquad \text{for } 300 \le L \le 350 \text{ m}$$
$$= 10.75 - [L/150 - 2.333]^{1.5} \qquad \text{for } 350 \le L \le 500 \text{ m}$$

Thus, from the sum of the bending moment in calm water and in waves (taking into account the sign, which assumes positive values in case of tensile stress on the deck)

$$M_{\text{tot}} = M_{\text{sw}} + M_{\text{w}} \tag{2.26}$$

it is concluded for the maximum bending stresses on the deck and bottom of the midship section:

$$\sigma_{1,2} = \text{M}_{\text{tot}} / W_{1,2} < 175 \text{N/mm}^2 \quad \text{(for common shipbuilding steel)} \tag{2.27}$$

where

$W_{1,2}$: section modulus $= I_M / z_{1,2}$
$z_{1,2}$: distances of deck and bottom respectively from the neutral axis
I_M : moment of inertia of midship section.

For the moment of inertia of midship section the following minimum value results as requirement:

$$\text{I}_{\text{M}} > 3CL^3 B(C_{\text{B}} + 0.7)(\text{cm}^4) \tag{2.28}$$

Effects of changing length Examining each of the aforementioned boundary loading conditions, for the ship moving in calm water or waves, we assume that the displacement Δ is fixed, as are the distribution of weights and buoyancy and the resultant loading curve of the ship's hull as a bending beam. Therefore, any changes in the bending moment $M(x)$ are simple functions of the length.

Considering an increase of the length by the ratio $\lambda = L_1/L_0$ (subscript 1: examined length, 0: original length), an increase of the bending moment by the same ratio is concluded:

$$M_1 = \lambda \cdot M_0 \tag{2.29}$$

Case A Assuming that for the ship under design, the displacement Δ, beam B, draft T, and the midship section (area and boundary profile) are fixed. The effects of a length change by the ratio λ are explored. Because of the fixed displacement, beam, and draft, it is clear that:

$$C_{\text{B1}} = (1/\lambda) C_{\text{B0}} \tag{2.30}$$

Provided that the midship section is fixed in terms of area and shape (profile), then the sectional modulus will remain unchanged, but the bending stresses will change by the ratio λ:

$$\sigma_1 = M_1 / W_1 = \lambda M_0 / W_0 = \lambda \cdot \sigma_0 \tag{2.31}$$

If we request that the stress level remains unchanged (similar construction material), namely:

$$\sigma_1 = \sigma_0$$

the moment of inertia must satisfy the following relation:

$$I_1 = \lambda I_0.$$

Considering for the simplified calculation of the moment of inertia of the midship section of the ship a *tubular* type bending beam of sectional area A_f, perimeter p, and average thickness t:

$$I = \kappa \cdot A_f \cdot d^2 = k \cdot p \cdot t \cdot d^2 \tag{2.32}$$

where

d = distance of the extreme structural points (the ship's deck or bottom) from the neutral axis,
κ = form coefficient of midship section.

It shows for constant midship section, that is, constant κ, p, and d, and constant ratio (I/t):

$$t_1 = \lambda \cdot t_0.$$

In conclusion, for maintaining the same level of bending stress it is *required to increase the average thickness t* by the ratio of lengths.

If the structural ship weight is expressed in the following form:

$$W_H = K_H \cdot A_H \cdot t \tag{2.33}$$

where

A_H: area of hull surface,
t: average plate thickness
K_H: form coefficient specific to midship section and ship type

it may be concluded for the ship under study:

$$W_{H1} = K_H \cdot A_H \cdot t_1 = \lambda \cdot K_H \cdot A_H \cdot t_0 \tag{2.34}$$

The area A_H can be approximated by Taylor's formula, which originally applies only to the hull's wetted surface, but can be herein extended for an assumed waterline at the deck height level:

$$A_{\mathrm{H}} = C_{\mathrm{H}} \cdot \sqrt{\nabla \cdot L} \tag{2.35}$$

where

$$C_{\mathrm{H}} = f(B/D, C_{\mathrm{MD}}) \tag{2.36}$$

However, it has been assumed that the midship section remains constant and the same applies to its area and perimeter, that is, B/D, $C_{\mathrm{MD}}=C_{\mathrm{M}}(T=D)$ and C_{H} constant. Thus, it is concluded for the hull surface:

$$A_{\mathrm{H1}} = C_{\mathrm{H}} \cdot \sqrt{\nabla \cdot L_1} = \sqrt{\lambda} A_{\mathrm{H0}} \tag{2.37}$$

and for the weights:

$$W_{\mathrm{H1}} = \lambda \cdot K_{\mathrm{H}} \cdot \sqrt{\lambda} \cdot A_{\mathrm{H0}} = \lambda^{3/2} W_{\mathrm{H0}} \tag{2.38}$$

In conclusion, if the displacement, beam, draft and midship section remain unchanged, *an elongation of the ship by the ratio of lengths* λ means an *increase* of the the ship's structural weight of the main hull (without superstructures) *by* $\lambda^{3/2}$ and a decrease of the block coefficient C_{B} by the ratio $(1/\lambda)$.

Case B: Assuming that the displacement Δ and the block coefficient C_{B} remain fixed, while the product $B \cdot T$ varies inversely proportional to the length, that is:

$$\Delta_1 = \Delta_0$$
$$C_{\mathrm{B1}} = C_{\mathrm{B_0}}$$
$$(BT)_1 = (1/\lambda)(BT)_0 .$$

It is also assumed that for small changes of the dimensions the specific form coefficient K_{H} for the calculation of the steel structure weight stays unchanged.

If it is required to maintain the same bending stress level $\sigma_1=\sigma_0$, then it is concluded:

$$M_1 \cdot (z_1 / I_1) = M_0 \cdot (z_0 / I_0) \tag{2.39}$$

where

$$\mathrm{I}_1 = k_1 \cdot p_1 \cdot t_1 \cdot d_1^2$$
$$(p_1 / p_0) \propto (B_1 / B_0) \propto (T_1 / T_0) \propto 1/\sqrt{\lambda}$$
$$(d_1 / d_0) \propto (T_1 / T_0) \propto 1/\sqrt{\lambda}$$
$$k_1 \cong k_0 .$$

Thus, we have for the moment of inertia:

$$I_1 = k_0 \cdot p_0 \cdot t_1 \cdot d_0^2 \cdot \lambda^{-3/2} \tag{2.40}$$

and due to

$$(z_1 / z_0) \propto (T_1 / T_0) \propto \lambda^{-1/2}$$
$$(M_1 / M_0) \propto \lambda$$

we obtain for the average plate thicknesses:

$$t_1 = \lambda^2 \cdot t_0 \tag{2.41}$$

Finally, substituting the last relationship into the equation of the steel structure weight of the main ship hull:

$$W_{\mathrm{H1}} = K_{\mathrm{H}} \cdot A_{\mathrm{H1}} \cdot t_1 \tag{2.42}$$

where

$$A_{\mathrm{H1}} = \lambda^{1/2} \cdot A_{\mathrm{H0}}$$

(for small changes of dimensions B, D and coefficient C_{MD}, see case A), it is concluded:

$$W_{\mathrm{H1}} = \lambda^{5/2} \cdot W_{\mathrm{H0}} \tag{2.43}$$

Thus, in the case that C_{B} is kept fixed during the length elongation with the ratio λ, the *increase of weight W_{H} is more drastic* ($\propto \lambda^{5/2}$) than in *case A* ($\propto \lambda^{3/2}$), where we had reduction of C_{B} by the ratio ($1/\lambda$) for fixed beam, draft and midship section.

Conclusions

1. As will be shown in Chap. 6, the cost of the steel structure of a ship is closely related to its weight. Therefore, a relatively high structural weight, as the result of an elongation at the expense of other characteristics of the vessel (see above elaborations), generally involves higher construction cost. Consequently, it is appropriate to keep the length as small as possible[7].
2. Besides the longitudinal strength, which was examined in this section and concerns the structural design of all ship types, the equally important *torsional stresses of open-deck ships*, such as containerships, LASH, etc., are also directly dependent on the length of the vessel.
3. The possible shift of displacement from the longitudinal direction (decrease of length) to the transverse (increase of beam) or vertical directions (increase of T) is beneficial for the longitudinal strength, but implies a shift of longitudinal strength problems to corresponding ones in the transverse direction; this requires special attention to the ship's structural design, but can be today readily addressed by modern FEM and other methods (see, for example shallow water, beamy large tankers) (Fig. 2.20).

[7] Experience says: *the smallest ship (least length) fulfilling shipowner's requirements is generally the best ("optimal").*

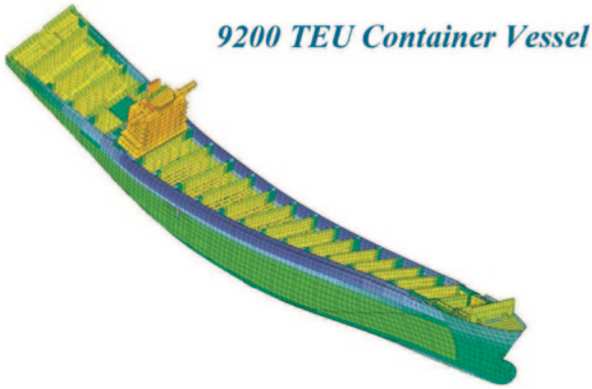


Fig. 2.20 Combined torsional and bending stresses on a containership (Study by Finite Element methods; Source: Germanischer Lloyd)

2.3.3 Effect of Length on the Outfitting Weight

As a general rule, increase of the length implies an increase of equipment and outfitting weight.

Examining the effects of an average increase of length on the various components of the equipment on board it is observed that it causes an

- Increase of the length of the piping systems (cargo, ballast, fire-fighting, etc.), cables, A/C airways, insulations, etc.
- Increase in the lateral profile area of the ship above waterline, resulting in an increase of the equipment number relevant to the ship's class, hence of the weight of anchors, chains, winches, etc.
- Increase of lateral profile area of the ship below waterline, resulting in increased demand for the ship's rudder area (the area ratio needs to remain constant, see Sect. 5.3).

It is assumed that the increase of the outfitting weight depends on the length ratio $\lambda = L_1/L_0$ with an exponent α_T:

$$W_{OT} \propto \lambda^{\alpha_T} \tag{2.44}$$

An increase of the weight W_{OT} generally implies an increase of the ship's construction cost (increased cost for materials and manhours for fitting).

2.3.4 Effect of Length on the Weight of Propulsion System and Fuel Consumption

As has been already detailed in Sect. 2.3.1, for ordinary ships with a Froude number $F_n \geq 0.15$, an increase of length generally leads to a reduction of the ship's total resistance for given speed and displacement. Due to the resulting reduction of the

required propulsion power, a decrease of the weight of propulsion installation and reduction of weight of carried fuel (for fixed service range) are concluded. This results in a reduced cost for purchasing the main engine, and a reduced operating cost in terms of the consumption of fuel, lubricant oil, etc.

2.3.5 Effect of Length on the Exploitation of Spaces and General Arrangement

The length of a cargo ship has a significant effect on the hold arrangements and the technique of the cargo-handling system. Thus, the number and length of the cargo holds, and the corresponding openings of the hatches, are directly related to the ship's length as well as to the size and location of the engine room.

Particular requirements concerning the configuration of hold spaces generally occur for heterogeneous cargoes, relating to the type, the form and size of break cargo, and to a lesser degree for homogeneous cargoes, namely for the mass bulk cargoes (dry or liquid) or for unitized cargoes.

The requirement for a specific number and size of holds or hatches, always relates to a minimum lower limit for the feasible length, while permitting the loading of various types of cargos and ensuring full holds.

Especially for bulk cargo carriers and particularly ore carriers, the requirement for an *odd* number (3, 5, 7, 9) of holds (so that it is possible to arrange an "alternate hold loading" due to strength and stability considerations) is an important factor for the length determination (Fig. 2.21).

Also, for ships carrying standardized large cargo units (unitized cargo), such as standard containers (ISO-Containers), barges, trailers, vehicles, and trains, namely containerships, LASH, Ro/Ro, car carriers/ train carriers, the relationship of the length to a multiple of the individual standard cargo length requires the selection of the ship's length within a certain limits, with little freedom to balance any differences to the desired length at the ends of the ship and the engine room.

Finally, for LNG-tanker, in the case of using spherical–LNG tanks, the hold length results from the ship's beam, due to the common tank diameter in both transverse and longitudinal directions and the given number of tanks in longitudinal direction (Fig. 2.22).

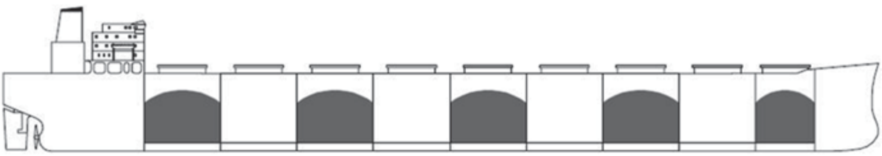


Fig. 2.21 Alternate hold loading for heavy ore cargo (fully loaded)

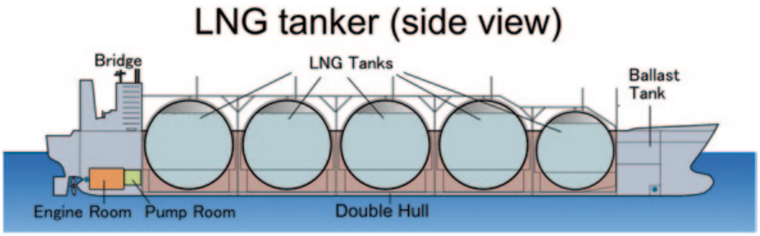


Fig. 2.22 LNG tanker (side view)

2.3.6 *Other Factors Affecting the Selection of Length*

1. Behavior in waves To avoid intense motions/accelerations in waves, which, beyond unfavorable structural loadings, lead to added resistance and additional powering in waves, thus also to *voluntary or involuntary* speed loss, regions of resonance of ship motions (of heave, pitch, and roll, which are characteristic by their natural periods/frequencies) should be avoided. In determining the ship's length, we are mainly interested in possible resonance in head seas, which primarily induce pitch and heave motions. Figure 2.23 (Lewis 1988) shows that, for a wavelength to ship length ratio $L_W/L = 1.0$ to 1.3 a resonance takes place and excessive values for both heave and pitch motions, which are mathematically coupled.

Of course, in practice it is difficult to avoid the resonance at certain wavelength, due to the existence of many wavelengths in the spectra of natural seas on earth, where seagoing merchant ships may operate. However, one may try to avoid resonance with the waves of higher energy density (at the *significant* wave period/length of known routes). These considerations are valuable for navigational areas for which sufficient statistical data of local wave spectra are available (especially for coastal ships) and they are anyway taken into account in naval ship design.

2. Freeboard The length significantly affects the freeboard of a ship, as it is the basis for calculating the *basic freeboard* in accordance with the Load Line Regulations (ICLL), see Sect. 2.19.

3. Passing limits of routes See dimensions of known canals/narrow straits, etc. (see Sect. 2.2).

2.3.7 *Ship Length Estimation Using Empirical Formulas*

Common empirical methods for estimating the length L are as follows:

a. Using coefficients ($L/\nabla^{1/3}$) for various ship types
b. Using semi-empirical mathematical formulas from statistical analyses that are based on purely economic criteria
c. Using semiempirical mathematical formulas derived from statistics of existing ships (based on hydrodynamic and economic criteria)
d. Using empirical diagrams for different types of ships

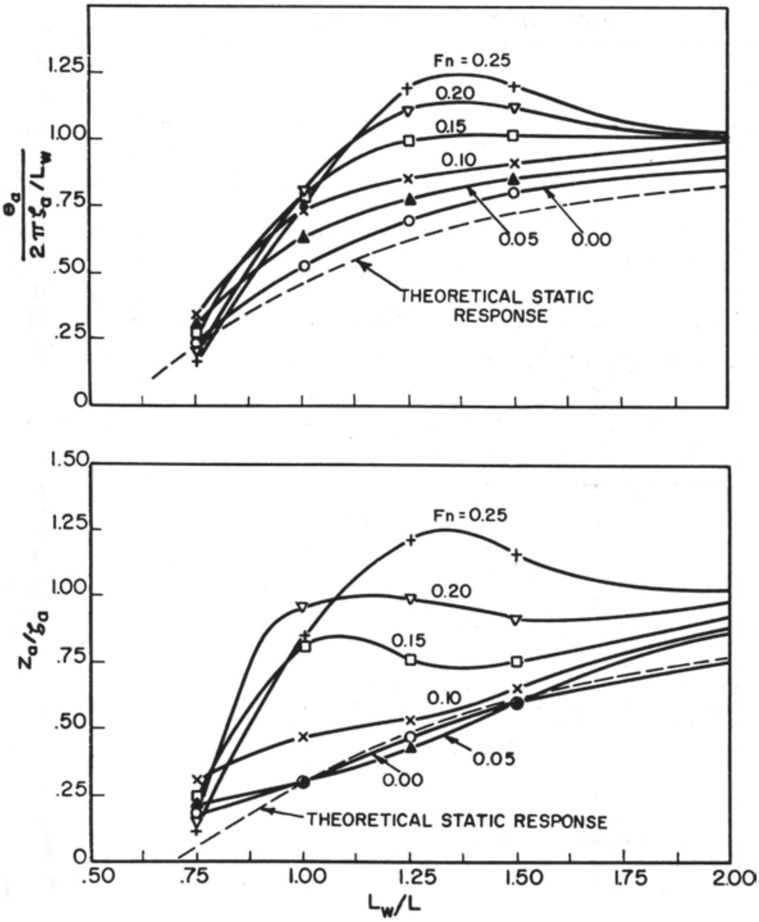


Fig. 2.23 Amplitude of pitch motion θ_α and heave motion z_α of a Series 60 ($C_B = 0.60$) model in head waves with amplitude ζ_α and length L_W, at different Froude numbers. (Lewis 1988)

Applications

a. After the prediction of the displacement and displaced volume ∇, it is possible to estimate the length by using the slenderness coefficients $L/\nabla^{1/3}$ from Tables 2.4 and 2.5 or from similar ships (see values in Appendix A).
b. Formula of "length of minimum building cost" according to Schneekluth (1985)

$$L = \Delta^{0.3} \cdot V^{0.3} \cdot C \tag{2.45a}$$

where

L: length between perpendiculars (m),
Δ: displacement (t),
V: service speed (kn) or

$$L = 1.22 \cdot \Delta^{0.3} \cdot V^{0.3} \cdot C \tag{2.45b}$$

for speed V in m/s

Table 2.4 Hull form coefficients and ratios of main dimensions for merchant ships (synthesis of original data by Strohbusch 1971, updated by Papanikolaou by use of IHS Fairplay World Shipping Encyclopedia, v. 12.01, 2011). Given upper and lower boundaries correspond to the standard deviation from the regression line of sample ships, as shown in Appendix A

Ship type	Hull form coefficients				Ratios of main dimensions		
	C_P	C_M	C_B	C_{WP}	L_{PP}/B	B/T	$L_{PP}/\nabla^{1/3}$
Fast seagoing cargo ships	0.57–0.65	0.97–0.98	0.56–0.64	0.68–0.74	5.7–7.8	2.2–2.6	5.6–5.9
Slow seagoing cargo ships	0.66–0.74	0.97–0.995	0.65–0.73	0.80–0.86	4.8–8.5	2.1–2.3	5.2–5.4
Coastal cargo ships	0.69–0.73	−0.985	0.58–0.72	0.78–0.83	4.5-5.5	2.5–2.7	4.2–4.8
Small short sea passenger ships	0.61–0.63	0.82–0.85	0.51–0.53	0.65–0.70	5.8-6.5	3.3–3.9	6.3–6.6
Ferries	0.53–0.62	0.91–0.98	0.50–0.60	0.69–0.81	5.9–6.2[a] 5.2–5.4[b]	3.7–4.0	6.2–6.9[a] 5.7–5.9[b]
Fishing vessels	0.61–0.63	0.87–0.90	0.53–0.56	0.76–0.79	5.1–6.1	2.3–2.6	5.0–5.4
Tugboats	0.61–0.68	0.75–0.85	0.50–0.58	0.79–0.84	3.8–4.5	2.4–2.6	4.0–4.6
Bulk carriers	0.79–0.84	0.990–0.997	0.72–0.86	0.88–0.92	5.0–7.1[a]	2.1–3.2	4.7–5.6
Tanker F_n =0.15	0.835–0.855	0.992–0.996	0.82–0.88	0.88–0.94	5.1–6.8	2.4–3.2	4.5–5.6
Tankers F_n =0.16−0.18	0.79–0.83	0.992–0.996	0.78–0.86	0.88–0.92	5.0–6.5	2.2–2.9	4.5–5.2
Fast seagoing reefers	(0.55)[c] 0.59–0.62	0.96–0.985	(0.53)[c] 0.57–0.59	0.68–0.72	6.7–7.2	2.8–3.0	6.1–6.5

[a] For $L > 100$ m
[b] For $L = 80-95$ m
[c] C_P, $C_B < 0.57$

Table 2.5 Hull form coefficients and ratios of main dimensions for merchant ships (synthesis of original data by Strohbusch, 1971, updated by use of IHS Fairplay World Shipping Encyclopedia, v. 12.01, 2011). Given upper and lower boundaries correspond to the standard deviation from the regression line of sample ships, as shown in Appendix A

Ship type	Ratio of main dimensions		
	L_{PP}/D	F_{FP}-$\%L_{PP}$	L_P-$\%L_{PP}$
Fast seagoing cargo ships	9.9–13.5	5.1–6.3	20–25
Slow seagoing cargo ships		5.8–7.0	30–35
Coastal cargo ships	10.0–12.0	up to 7.0	40–50
Small short sea passenger ships	10.4–11.6	6.6–7.9	20–25
Ferries	8.6–10.3	7.0–10.0	25–35
Fishing vessels	8.2–9.0	8.0–8.5	15–25
Tugboats	7.7–10.0	8.2–10.2	20–30
Bulk carriers	10.5–12.8	4.4–4.9	50–60
Tankers F_n =0.15	12.0–14.0	3.6–4.5	50–60
Tankers F_n =0.16−0.18	10.5–12.8	4.4–4.9	50–60
Fast seagoing reefers	−11.0	5.6–6.6	10–15

For both cases C takes the following value:

$$C = 3.2 \qquad \text{for } C_B = 0.145/F_n$$
$$= 3.2\frac{C_B + 0.5}{(0.145/F_n) + 0.5} \qquad \text{for } C_B \neq 0.145/F_n$$

The above constraints in the formula for the C_B are understood approximately.

The basic limitations for applying the above empirical formula are as following:

1. $\Delta \geq 1{,}000$ t.
2. V corresponding to $0.16 \leq F_n \leq 0.32$.
3. C_B within the boundaries $0.48 \leq C_B \leq 0.85$.
4. Proportional correction of the constant C (increase) for restrictions on B and T and high ratio of volume below D to displaced volume (∇_D/∇).
5. Correction of constant C (decrease) for the existence of optimized bulbous bow.

The constant C can be alternatively calculated by using the following formula (Friis et al. 2002):

$$C = 3.4 - (\Delta - 10^3)/10^6 \qquad \text{for} \quad 1{,}000\ \text{t} \leq \Delta \leq 201{,}000\ \text{t}$$
$$= 3.2 \qquad \text{for} \quad \Delta > 201{,}000\ \text{t}$$

The above formula by Schneekluth (1985) is the result of statistical analysis of data of optimized ships with respect to only construction cost. However, taking into account as well the operating cost, which is equally important for the owner's interests, an increase of about 10 % of the length resulting from the above formula is recommended (which leads to lower resistance, reduced propulsive power and fuel cost).

c. Formulas from statistical analyses of data of existing ships[8]

1. Ayre's formula for length estimation:

$$L_{PP}/\nabla^{1/3} = 3.33 + 1.67\ V/\sqrt{L_{PP}} \tag{2.46}$$

2. Posdunine/V. Lammeren's formula for length estimation

$$L_{WL}/\nabla^{1/3} = CV/(V+2)^2 \tag{2.47}$$

where

C = 7.62 (all types, Posdunine)
= 7.16 (cargo ships, V. Lammeren)
= 7.32 (fast twin-screw ships, V. Lammeren)
= 7.92 (fast passenger ships, V. Lammeren)

3. Völker's formula for length estimation

$$L_{PP}/\nabla^{1/3} = C_1 + 4.5V/\sqrt{g \cdot \nabla^{1/3}} \tag{2.48}$$

[8] All below formulas refer to the data of old ships; they deliver in general larger lengths than used today in practice; they are, however, a good yardstick for evaluating possible ship lengths at the conceptual design stage.

where

C_1 = 3.5 for dry bulk cargo ships/containerships
= 3.0 for reefer ships
= 2.0 for fishing/short sea cargo ships.

Notes on units in formulas 1 to 3 (Eqs. 2.46, 2.47, 2.48):

1. L (m); V (kn)
2. ∇: displaced volume (m^3); V (kn)
3. g (m/s^2): gravitational acceleration; V (m/s): design speed (service)

d. Use of diagrams for various types of ships

1. Figure 2.24: Relation of L_{PP} and of $L \cdot B \cdot D$ to the required hold capacity for tankers (Lamb 2003).
2. Figure 2.25: Relation of the ratios L_{PP}/B and L_{PP}/D to the required hold capacity for tankers (Lamb 2003).
3. Figure 2.26: Relation of the L_{PP} (B and D) to the required hold capacity ∇_{REQ} for cargo ships according to Watson and Gilfillan (1976).

Instructions for use graph 2.26

3.1. Estimation of ∇_{REQ} based on the required capacity GRAIN (+1 to + 2%) or BALE (+11 to +12%), for example 20,000 m^3.
3.2. Assumption of L/B and B/D based on similar ships, for example $L/B = 6.5$ and $B/D = 1.8$.
3.3. Assume the engine room position to be abaft or 3/4 of length abaft; here, for example, 3/4 L abaft amidships.
3.4. Find hull's total volume below the main deck (abscissa) ∇_H, for example 27,560 m^3, and the corresponding engine room volume, $\nabla_M = \nabla_H - \nabla_R$, for example, 7,560 m^3.
3.5. Find the product of $(L \times B \times D)$ (ordinate), for example 38,760 m^3, based on the approximation of the block coefficient C_{BD}at the height of D, for example 0.70. The latter may be estimated based on C_B, for the draft T, see 2.9, and use of the side graph of Fig. 2.26 (bottom left of Fig. 2.26).
3.6. Find the main dimensions of L_{PP}, B, and D from the side graph of Fig. 2.26 (top left), for example $L_{PP} = 139.8$ m, $B = 21.5$ m, $D = 12$ m, where the straight lines $L/B = 6.5$ and $B/D = 1.8$ can be replaced with other values, which correspond to respective similar ships.

4. Figure 2.27: Approximations of L_{PP}, coefficient $L \cdot B \cdot D$ and the ratio L_{PP}/B for ships carrying standardized containers as a function of the total number of transported TEU containers (Twenty Feet Equivalent Unit ISO standardized boxes of 20 ft length, 8 ft breadth and 8 (8.5) ft height).

5. Figure 2.28: Relation of the displacement Δ to the length L_{PP} (a), the coefficient $L \cdot B \cdot D$ to the installed power (b), and the ratio L/D to the waterline length L_{WL} (c) for *tugboats* (Lamb 2003).
6. Figure 2.29: Relation of the main dimensions and other characteristics for North Sea *fishing vessels* to the volume of fish hold (refrigerated hold) (Henschke 1964).

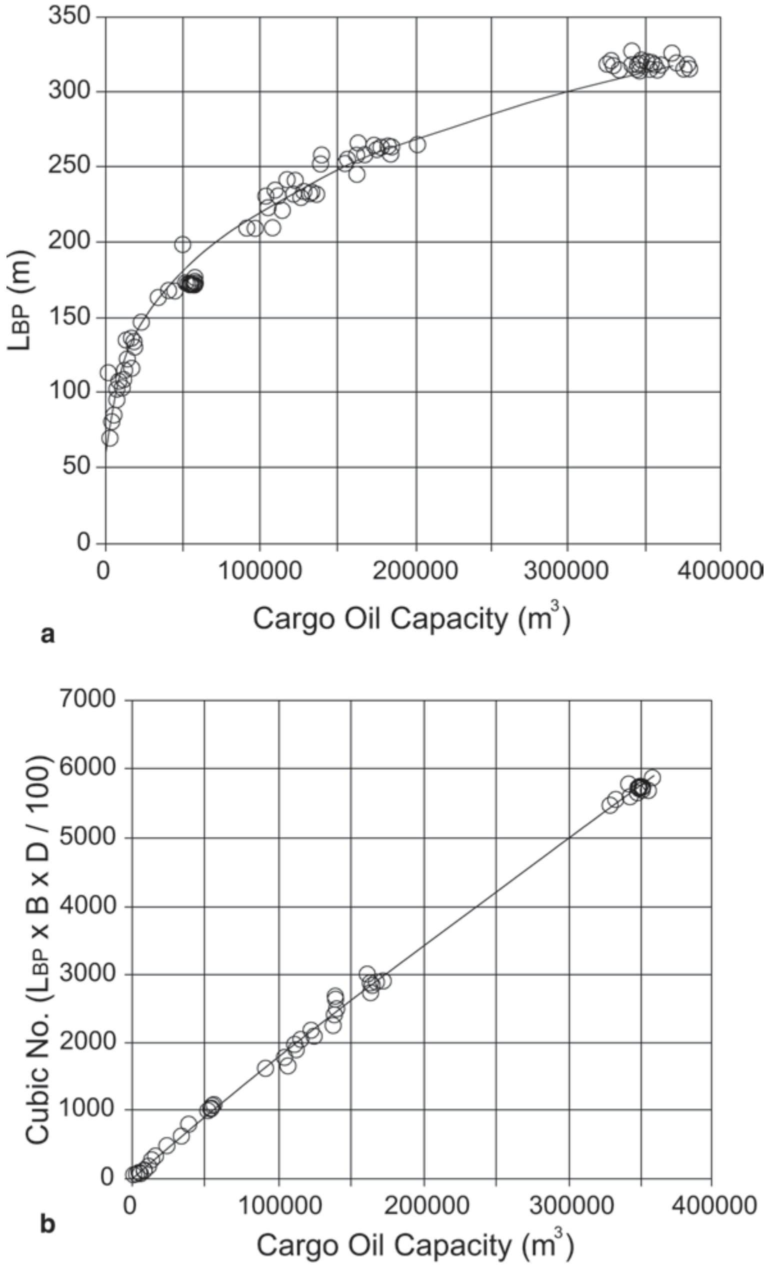


Fig. 2.24 Relations of L_{PP} and volumetric numeral $L \times B \times D$ to the hold capacity for tankers. (Lamb 2003)

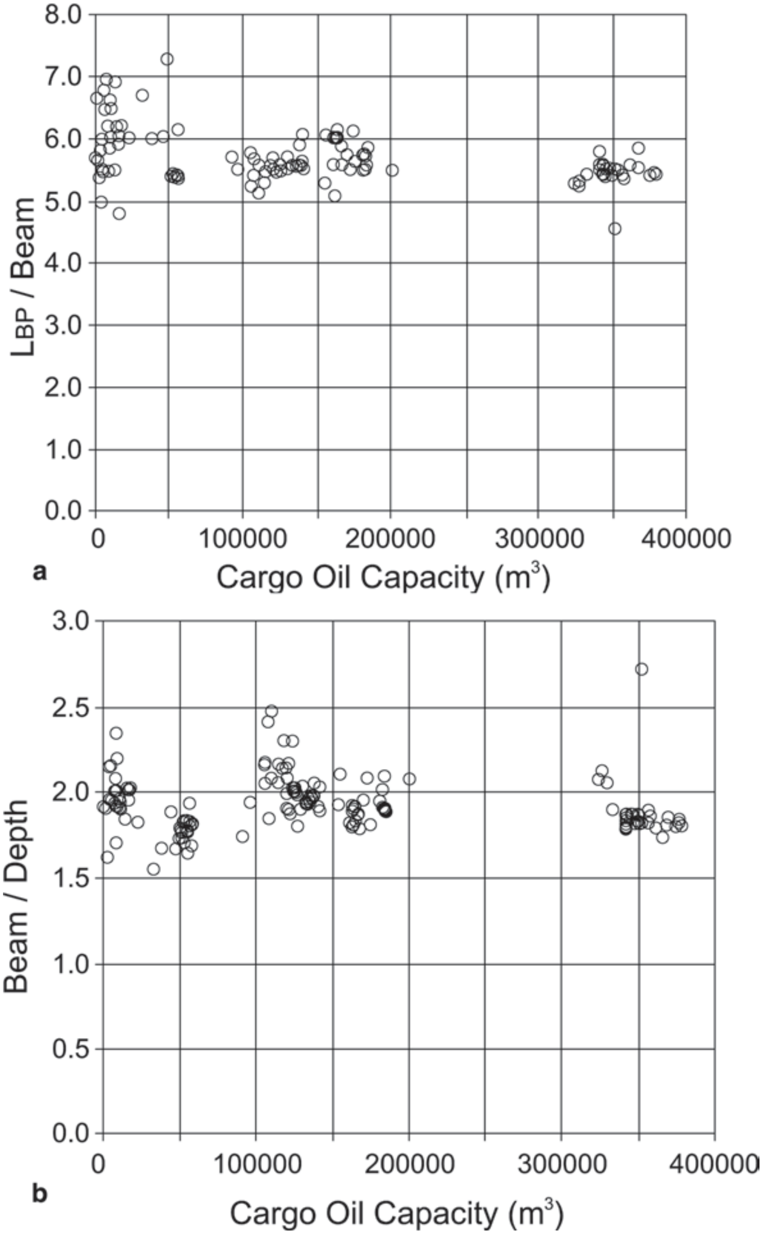


Fig. 2.25 Relation of the ratios L_{pp}/B. **a** and L_{pp}/D. **b** to the hold capacity for tankers. (Lamb 2003)

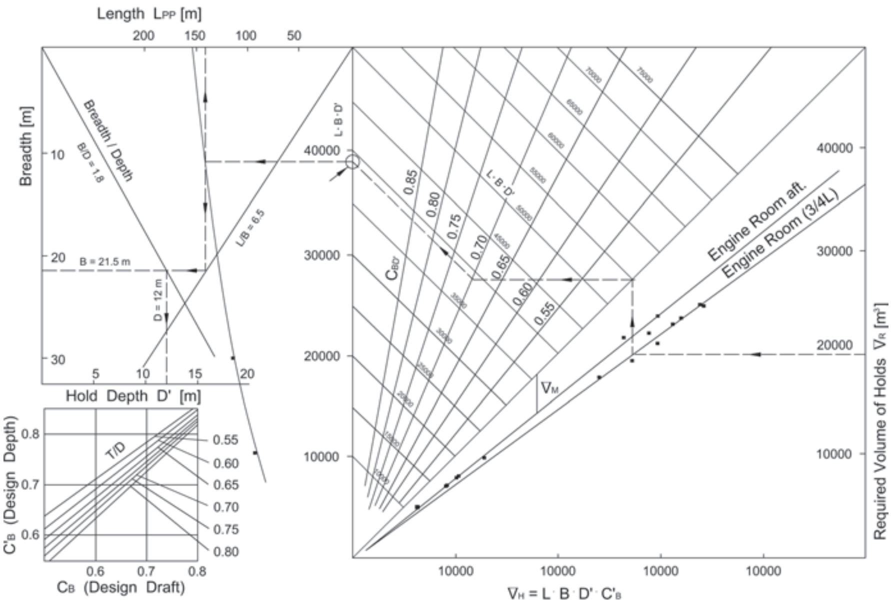
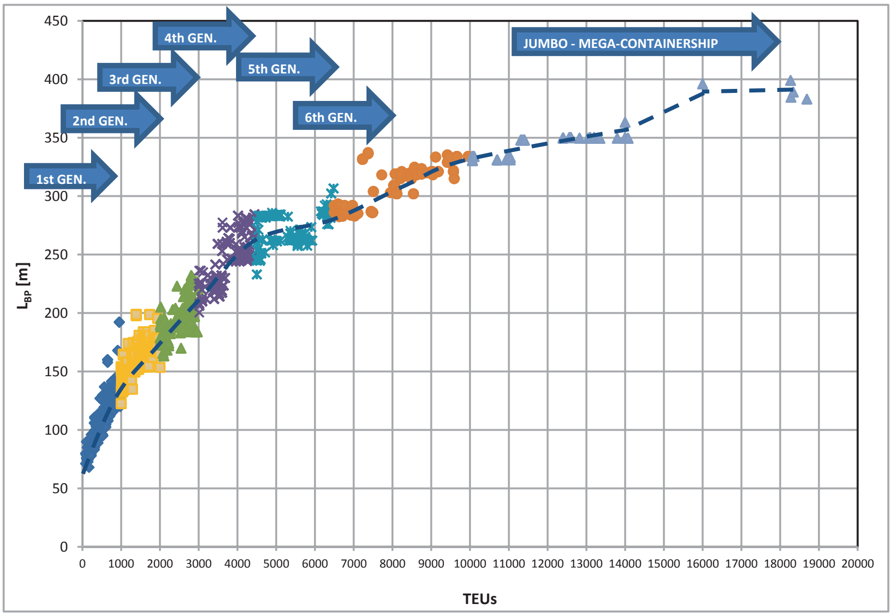
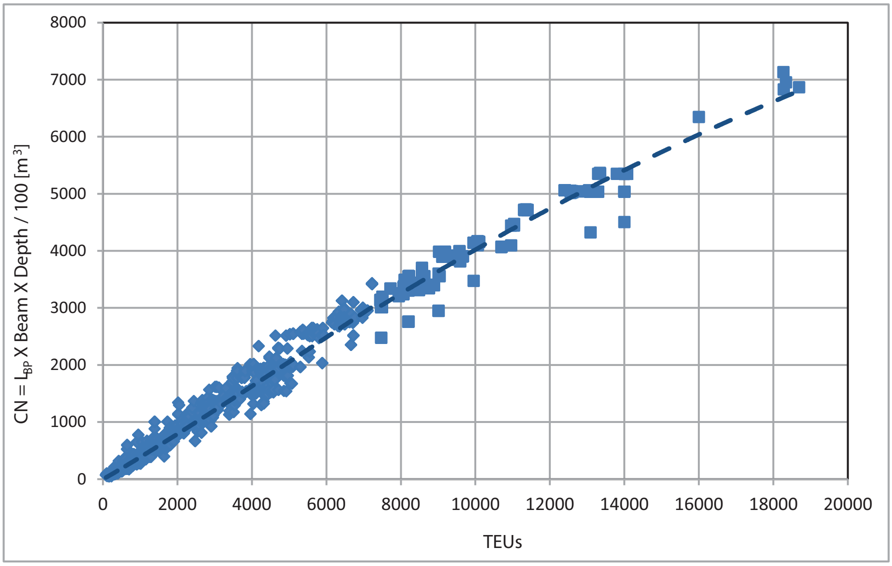


Fig. 2.26 Determination of main dimensions based on the required hold capacity under main deck ∇_R according to Watson and Gilfillan (1976), for $L/B = 6.5$ and $B/D = 1.8$

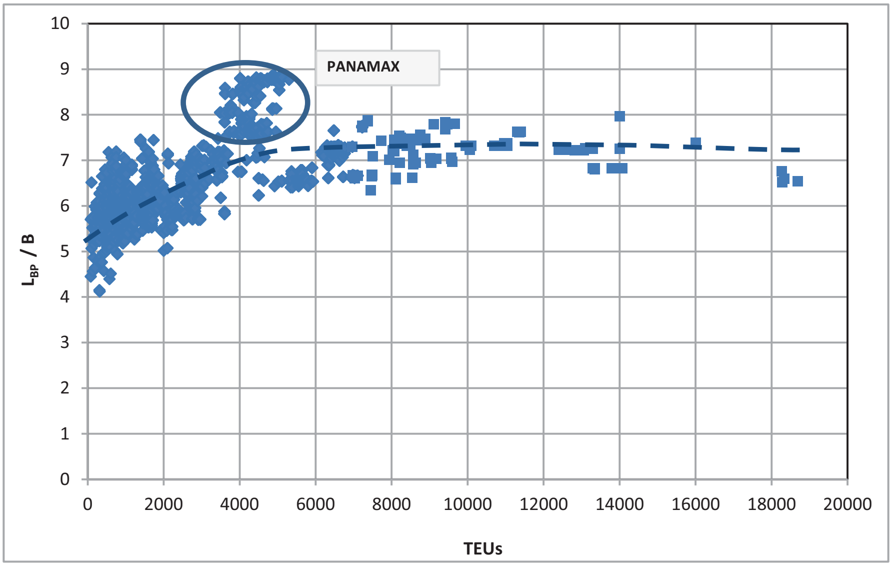


a

Fig. 2.27 Relations of (a) length L_{PP}, (b) volumetric numeral $L{\cdot}B{\cdot}D$ and (c) the ratio L_{PP}/B to the total number of transported TEU containers for containerships. (Papanikolaou 2014)



b



c

Fig. 2.27 (continued)

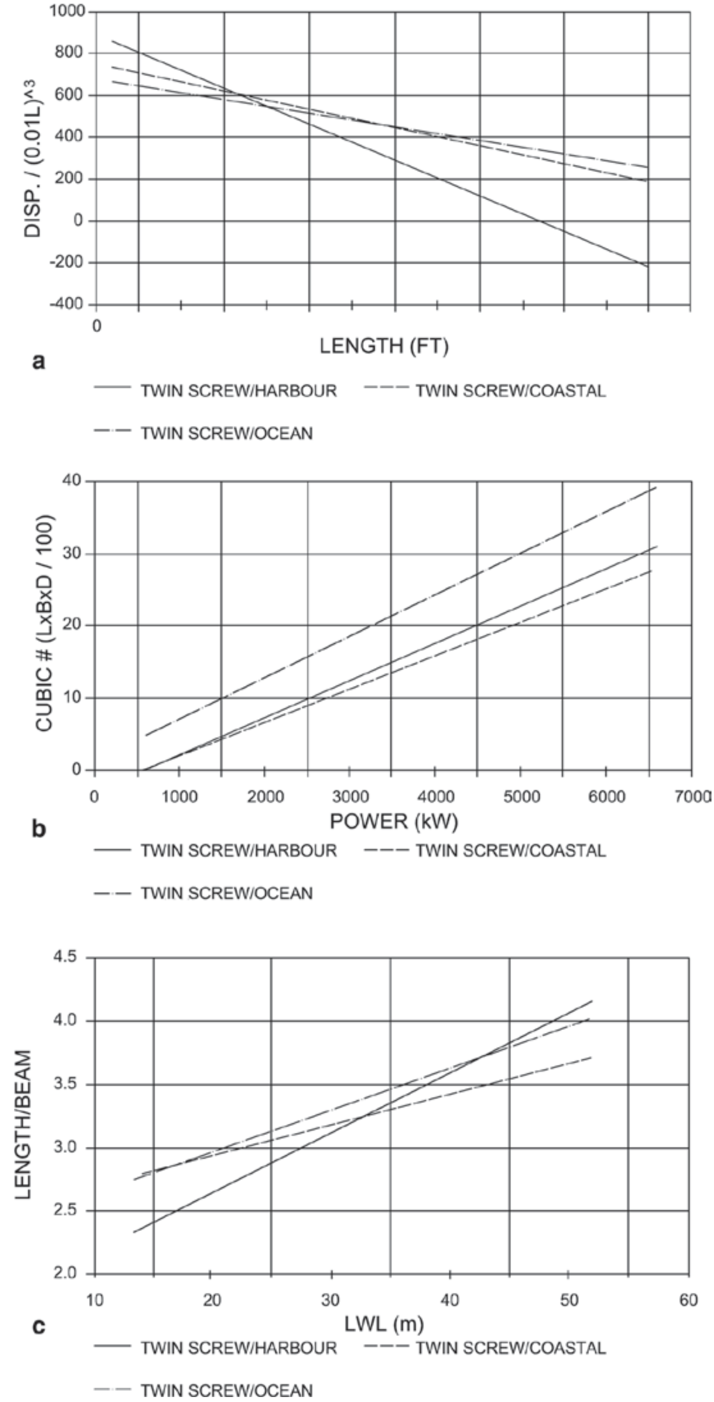


Fig. 2.28 (a) Relation of displacement with the length L_{PP}, (b) the volumetric numeral $L{\cdot}B{\cdot}D$ with the installed power and (c) the ratio L/B with the waterline length L_{WL} for tugboats. (Lamb 2003)

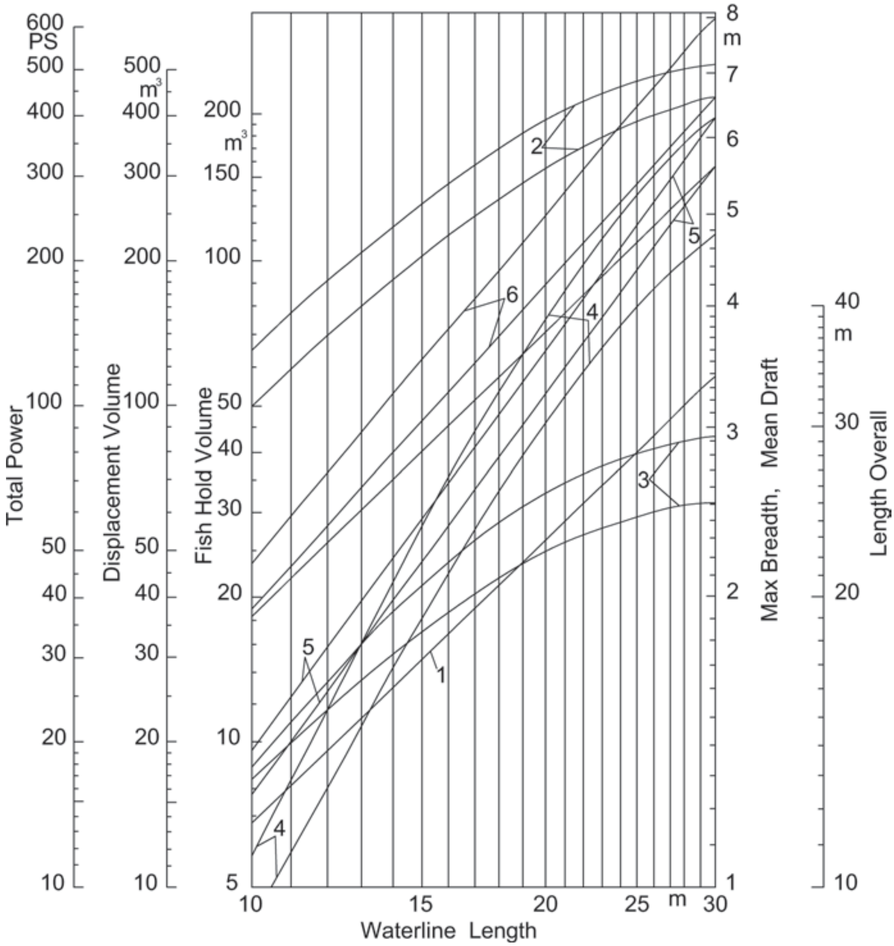


Fig. 2.29 Relationships of the length L_{WL} to other dimensions and basic characteristics for North Sea fishing vessels. (Henschke 1964). (*1*. Overall length, *2*. Beam (maximum), *3*. Average draft, *4*. Fish hold volume, *5*. Displaced volume, *6*. Installed engine power)

7. Figure 2.30: Relation of L_{PP}, B, and D to the refrigerated hold capacity for fishing ships (Lamb 2003).
8. Figure 2.31: Relation of L_{OA}, B, and T to deadweight for Chemical Tankers (Lamb 2003).
9. Figure 2.32: Statistical averages of slenderness coefficients of oceangoing ships according to Völker (1974).

e. Recommended procedure for the determination of length

   e1. Approximation of L based on the slenderness coefficient (see procedures (a) and (c))

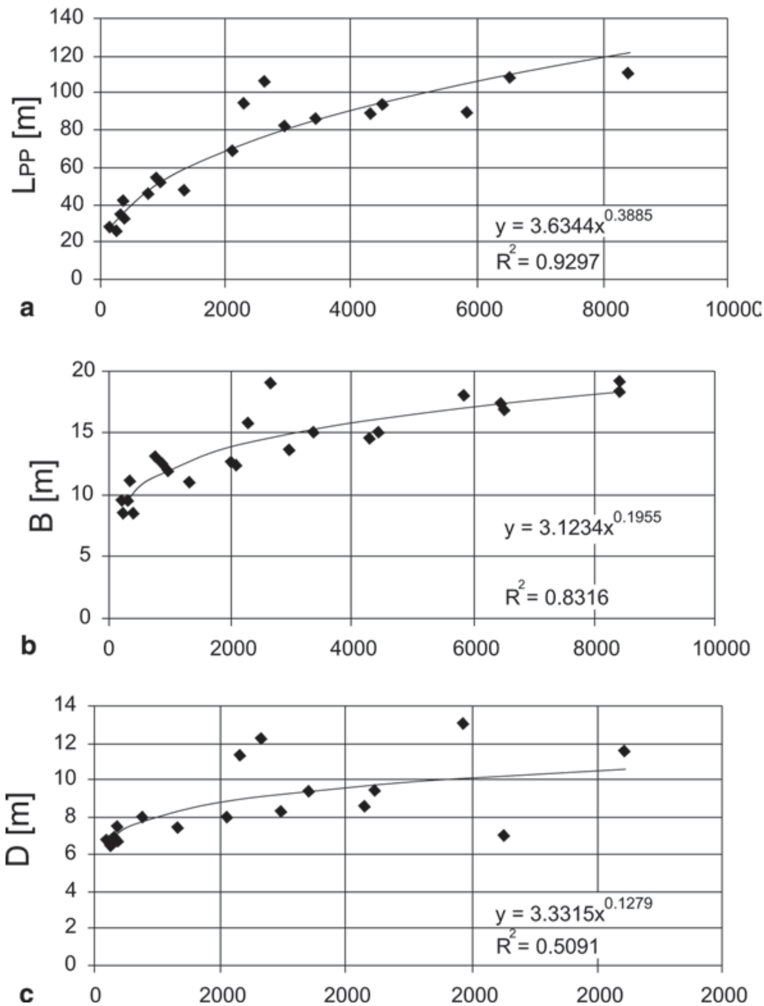


Fig. 2.30 Relationships of length L_{PP}, beam B, and side depth D to refrigerated hold capacity for fishing vessels. (Lamb 2003). (length (m) (**a**), beam (m) (**b**), side depth (m) (**c**))

e2. Examination of the resultant L based on the "least cost" formula according to Schneekluth (b)

e3. Examination of the resultant L based on the empirical diagrams (d)

e4. Examination and adjustment of L with regard to the physical, passing constraints: physical limits of channels, canals, ports, slipways, or docks of the shipyards

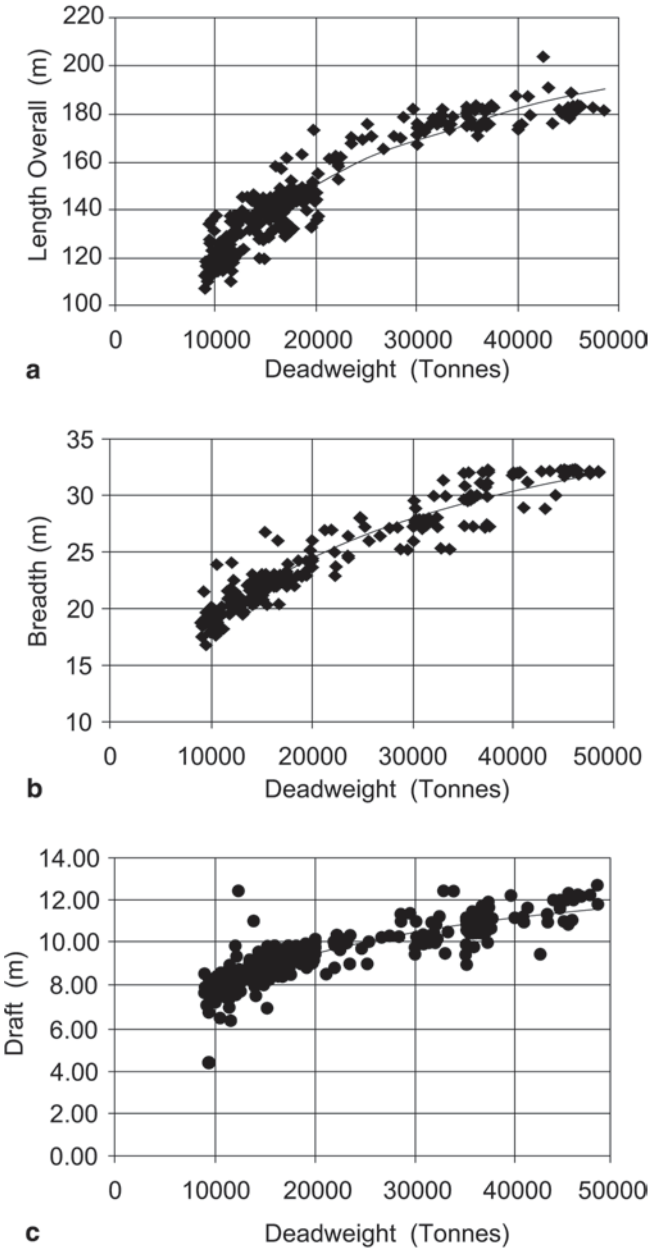


Fig. 2.31 Relationships of length L_{OA} (**a**), beam B (**b**), and draft T (**c**) to deadweight for chemical tankers. (Lamb 2003)

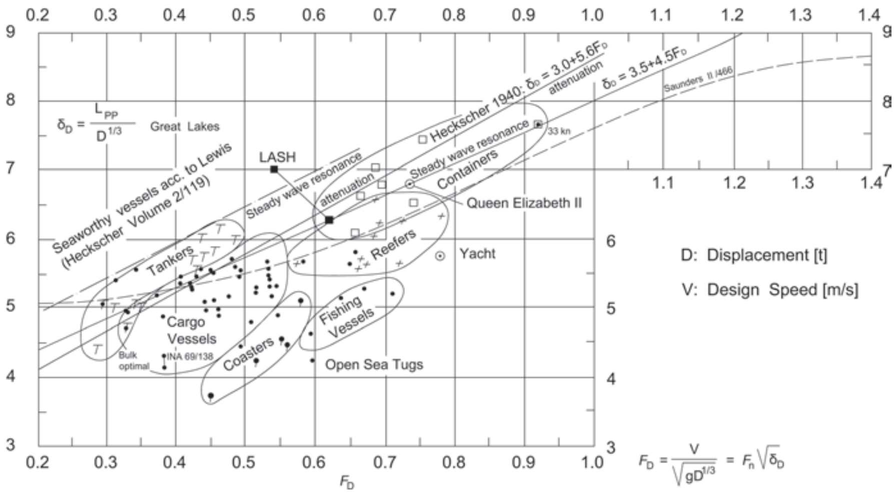


Fig. 2.32 Average slenderness coefficients of ocean-going ships according to Völker (1974)

e5. Examination of L with respect to the required number of transverse bulkheads according to the specifications of a recognized classification society[9]; possible adjustment of the length in cases of marginal exceedance of the limit for certain required number of bulkheads (bulkhead/steel weight savings)

e6. Examination of L, in conjunction with side depth D, regarding the ratio of (L/D) that needs to be below certain limit according to the rules of specific classification society

e7. Examination of L with respect to the possible occurrence of resonance of ship motions in typical waves in the region of operation (to avoid $\lambda_W \sim L$); this only applies to vessels with special requirements in terms of seakeeping, such as passenger ships and naval ships in general

e8. Examination of L with respect to the superposition of the generated bow wave, stern wave and shoulder waves for certain speeds of the ship due to possible excessive increase of wave resistance; indirectly, examination of the appropriateness of the operational Froude number

In the preliminary design stage, the above process is limited to the first six steps only (1–6).

[9] Every ship must have at least one collision bulkhead, one after peak bulkhead, and one bulkhead at the fore and aft boundaries of the engine room. In case the engine room is placed astern, the after-peak bulkhead coincides with the aft bulkhead of the engine room. The total number of bulkheads as a function of ship's length L in accordance with the regulations of, for example, Lloyd's Register is as follows:

$L \le 65$ m, $N=3$ (4); 65 m$<L\le 85$ m, $N=4$ (4); 85 m$<L\le 90$ m, $N=5$ (5); 90 m$<L\le 105$ m, $N=5$ (5); 105 m$<L\le 115$ m, $N=5$ (6); 115 m$<L\le 125$ m, $N=6$ (6); 125 m$<L\le 145$ m, $N=6$ (7); 145 m$<L\le 165$ m, $N=7$ (8); 165 m$<L\le 190$ m, $N=8$ (9); $L>190$ m, N as appropriate.

The above applies to ships with engine rooms placed astern (in parenthesis the corresponding number of bulkheads for the engine room placed amidships).

2.4 Slenderness Coefficient $L/\nabla^{1/3}$

The Slenderness or sharpness coefficient $L/\nabla^{1/3}$ or the inverse of this value's third power ∇/L^3 which is often referred to as volumetric coefficient (and is preferred in Anglo-Saxon countries), expresses the hull slenderness, especially in combination with the prismatic coefficient *C*P. Generally, high values of slenderness coefficient and low C_p values imply fine-lined hulls generally for fast ships (Fig. 2.33).

2.4.1 Influence on the Ship's Resistance

The influence of the slenderness coefficient on the ship's wave resistance is obvious, especially for fast ships. This can be easily concluded both from the phenomenological point of view (see Sect. 2.3.1), and practically from the application point of view, namely when using well known semiempirical formulas for calculating the residuary resistance, for example according to the method of Guldhammer (FORMDATA), where the slenderness ratio is a basic parameter.

For fast ships, a high slenderness coefficient leads to a reduction of the *intensity* of the generated, ship-bound waves, and consequently of the wave resistance; generally, it contributes to a reduction of the ship's *form* (or residuary) resistance, thus beyond the wave-making resistance also of the pressure viscous resistance.

For relatively slow ships, with low wave resistance percentage values, the requirement for a least wetted surface for given displacement (what minimizes the frictional resistance) leads to a length that is as small as possible and results in ships with small lengths, comparably large beams and drafts, as well as high fullness, that is, high block coefficients and relatively low slenderness coefficients.

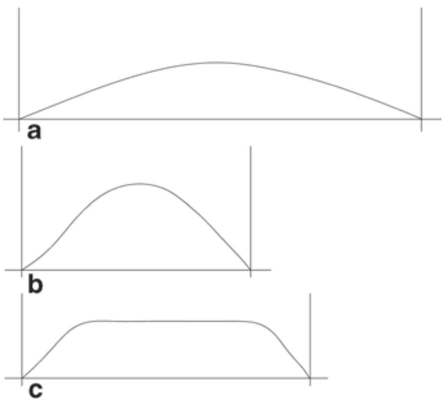


Fig. 2.33 Examples of effect of slenderness ratio and prismatic coefficient on hull form. (**a**) Fine-lined hull form, high $L_{PP}/\nabla^{1/3}$ and low C_P, typical for fast ocean liners, naval ships etc. (**b**) Short and sharp at the ends hull form, low $L_{PP}/\nabla^{1/3}$ and low C_P, typical for fishing and offshore support vessels, etc. (**c**) Full hull form, high $L_{PP}/\nabla^{1/3}$, and high C_P, typical for slow cargo ships, bulkcarriers, tankers etc.

2.4.2 Effect on the Ship's Structure

In consistency with the effort to minimize the frictional resistance for relatively slow ships with the distribution of displacement over a relatively short length, large beam, and draft (hence also of side depth), it is concluded that low slenderness coefficients combined with high block coefficients, lead to relatively simple and economical structures. The increased distribution of displacement in the transverse direction may be limited in extreme situations by transverse strength problems that require special transversal strengthening.

2.4.3 Approximate Values

Approximate values of the slenderness coefficient for common ship types are given in Tables 2.6 and 2.7 (and in Appendix A).

In the preliminary design phase and especially for deadweight carriers (see Sect. 1.4.2), it is appropriate to preliminarily estimate the length through the slenderness coefficient of similar ships. The resulting length can be examined by well-established empirical or semiempirical formulas (see Sect. 2.3).

2.5 Selection of Other Main Dimensions

After the preliminary estimation of the ship's length (see Sect. 2.3) and of the block coefficient C_B (see more details in Sect. 2.10), as well as of the displacement (see Sect. 2.1) (in the case of deadweight carriers), we commonly proceed with the selection of the beam B and draft T, which are directly related to each other, namely through

$$B \cdot T = \nabla / (L \cdot C_B). \tag{2.49}$$

The basic factor that influences the selection of B and T is at first possible topological limits of the route, that is restrictions on the beam in terms of the passage of canals and channels, for example for Panamax ships (passing through the Panama Canal, $B_{max} = 32.31$ m/106 ft). Also, there may be limitations for the ship's operational draft due to the ship's approach to river estuaries, transiting through canals or channels, calling at certain ports of limited depth (for example for Panamax ships, $T_{max} = 12.09$ m or 39 ft, 6 in) .

The *minimum* values for the beam are determined by the requirements for adequate stability, while for the draft the main requirement arises from the need of fitting a propeller of as large as possible diameter (for achieving higher efficiency). This applies particularly to ships of increased towing power (like tugboats and fishing vessels).

Table 2.6 Hull form coefficients and ratios of main dimensions for merchant ships (synthesis of original data by Strohbusch, 1971, updated by Papanikolaou by use of IHS Fairplay World Shipping Encyclopedia, v. 12.01, 2011). Given upper and lower boundaries correspond to the standard deviation from the regression relationship of sample ships, as shown in Appendix A.

Ship type	Hull form coefficients				Ratios of main dimensions		
	C_P	C_M	C_B	C_{WP}	L/B	B/T	$L_{PP}/\nabla^{1/3}$
Fast seagoing cargo ships	0.57–0.65	0.97–0.98	0.56–0.64	0.68–0.74	5.7–7.8	2.2–2.6	5.6–5.9
Slow seagoing cargo ships	0.66–0.74	0.97–0.995	0.65–0.73	0.80–0.86	4.8–8.5	2.1–2.3	5.2–5.4
Coastal cargo ships	0.69–0.73	-0.985	0.58–0.72	0.78–0.83	4.5–5.5	2.5–2.7	4.2–4.8
Small short sea passenger ships	0.61–0.63	0.82–0.85	0.51–0.53	0.65–0.70	5.8–6.5	3.3–3.9	6.3–6.6
Ferries	0.53–0.62	0.91–0.98	0.50–0.60	0.69–0.81	5.9–6.2[a] 5.2–5.4[b]	3.7–4.0	6.2–6.9[a] 5.7–5.9[b]
Fishing vessels	0.61–0.63	0.87–0.90	0.53–0.56	0.76–0.79	5.1–6.1	2.3–2.6	5.0–5.4
Tugboats	0.61–0.68	0.75–0.85	0.50–0.58	0.79–0.84	3.8–4.5	2.4–2.6	4.0–4.6
Bulk carriers	0.79–0.84	0.990–0.997	0.72–0.86	0.88–0.92	5.0–7.1[a]	2.1–3.2	4.7–5.6
Tankers F_n=0.15	0.835–0.855	0.992–0.996	0.82–0.88	0.88–0.94	5.1–6.8	2.4–3.2	4.5–5.6
Tankers F_n=0.16–0.18	0.79–0.83	0.992–0.996	0.78–0.86	0.88–0.92	5.0–6.5	2.2–2.9	4.5–5.2
Fast seagoing reefers	(0.55)[c] 0.59–0.62	0.96–0.985	(0.53)[c] 0.57–0.59	0.68–0.72	6.7–7.2	2.8–3.0	6.1–6.5

[a] For $L > 100$ m
[b] For $L = 80–95$ m
[c] Rarely: C_P, $C_B < 0.57$

Regarding the influence of *B/T* ratio on resistance, the frictional resistance, which is directly related to the wetted surface of the hull, is minimized for a *B/T* value around 2.5 and approximately the same applies to the residuary resistance, if there are no other restrictions or requirements on the absolute *B* and *T* values.

Thus the *B/T* ratio is usually selected close to 2.5 and possible exceedances are usually due to restrictions relating to limitations of the draft (always occurring for large tankers and bulk carriers) or due to particular, enhanced requirements on stability (for example for ROPAX ships). Note that significantly smaller values than 2.5 are rare.

The beam can be determined based on the *L/B* ratio of similar ships (see Table 2.6) and following this the draft *T* can be approximated through the chosen

Table 2.7 Hull form coefficients and ratios of main dimensions for merchant ships (synthesis of original data by Strohbusch, 1971, updated by use of IHS Fairplay World Shipping Encyclopedia, v. 12.01, 2011). Given upper and lower boundaries correspond to the standard deviation from the regres-sion line of sample ships, as shown in Appendix A

Ship type	Ratio of main dimensions		
	L_{PP}/D	F_{FP}-% L_{PP}	L_P-% L_{PP}
Fast seagoing cargo ships	9.9–13.5	5.1–6.3	20–25
Slow seagoing cargo ships		5.8–7.0	30–35
Coastal cargo ships	10.0–12.0	up to 7.0	40–50
Small short sea passenger ships	10.4–11.6	6.6–7.9	20–25
Ferries	8.6–10.3	7.0–10.0	25–35
Fishing vessels	8.2–9.0	8.0–8.5	15–25
Tugboats	7.7–10.0	8.2–10.2	20–30
Bulk carriers	10.5–12.8	4.4–4.9	50–60
Tankers $F_n = 0.15$	12.0–14.0	3.6–4.5	50–60
Tankers $F_n = 0.16$–0.18	10.5–12.8	4.4–4.9	50–60
Fast seagoing reefers	−11.0	5.6–6.6	10–15

B/T ratio. The influence of *L/B* on the ship's resistance is not straightforward, like that of the slenderness coefficient $L/\nabla^{1/3}$, though one would generally expect that a lower *L/B* ratio affects negatively ship's wave resistance. However, for a given draft *T*, length *L* and displacement, an *increase* of beam *B*, or *reduction* of the ratio *L/B*, means *reduction* of the block coefficient $C_B$and consequently *possible reduction* of the total resistance (see example, see Sect. 2.6.2).

The above considerations apply mainly to "normal" general dry cargo ships or liquid cargo ships without special requirements in terms of the transported cargo type or stability. However, for cargo ships transporting standardized/unitized cargos of fixed size (*linear dimension ships*) , for example containerships, Ro-Ro, etc., the beam generally changes stepwise, depending on the number of transversely stowed standardized (unitized) cargo, for example for *containerships of about Panamax size*:

$$B \cong 3n + 2.2\text{m} \tag{2.50}$$

where n is the number of transversely stackable standardized containers under deck (TEU containers; standard cross-section in feet: $8' \times 8'$ and up to $8.0' \times 8.5'$).

The beam's influence on stability, especially on the initial stability (metacentric height $\overline{GM}$) is drastic, given that a small increase of beam leads to significant increase of $\overline{BM}$ (see Sect. 2.6).

Regarding the selection of draft, the factors that have significant influence are briefly listed as follows:

- Large draft contributes to the selection of propellers of higher efficiency due to the possible fitting of a large diameter propeller (low thrust/load coefficient) and low number of propeller revolutions (rpm); it allows, also the fitting of larger rudders for improved maneuverability.

- Large draft requires strengthening of the ship's structural elements in the bottom area and lower hull shell.
- The resulting *freeboard* of the ship, defined as the difference between the selected draft and the upper side of the bulkhead deck (side depth D), must be in any case *greater* than the resultant *minimum* freeboard value derived from application of the International Load Line Convention regulations.

As to stability, the influence of an *increase* of draft is complicated and is certainly associated with possible changes of other dimensions, that is, of the ship's length and in particular the ship's beam:

- If other ship sizes (such as L, B, and water plane area) are assumed fixed, but the displacement increases (*due to the draft increase*), then the metacentric radius $\overline{BM}$ will decrease. The same will happen even more drastically, if for a given displacement and length, the beam of the ship decreases in parallel to the increase of the draft (in order to keep the block coefficient unchanged).
- If the side depth remains fixed and the freeboard is at acceptable level, the maximum value and the range of the righting arm *will decrease* due to premature immersion of the deck edge into water. For certain hulls, where the immersion of the deck follows the emergence of the bottom, just the opposite may happen.
- An increase of the distance of the center of buoyancy from the base leads to increased $\overline{KB}$; thus, a possible reduction of $\overline{BM}$ may be partially balanced by the increase of $\overline{KB}$ resulting in an increase or decrease of $\overline{KM}$ depending on the hull form.

A large draft may be excluded due to topological limiting requirements of routes.

A useful formula for the selection of B and T through the ratio (L/B) is concluded from an algebraic processing of the definition of C_B:

$$\nabla = L \cdot B \cdot T \cdot C_\mathrm{B} = L^3 \cdot C_\mathrm{B} / [(L/B)^2 \cdot B/T] \Rightarrow B/T = \frac{L^3 C_\mathrm{B}}{(L/B)^2 \nabla} \tag{2.51}$$

which in combination with the equation

$$B \cdot T = \frac{\nabla}{L \cdot C_\mathrm{B}} \tag{2.52}$$

leads to the values for B and T (two equations for two unknowns).

The effect of changing B and T by δB and δT, respectively on the stability can simply be examined on the basis of the resulting changes of the metacentric radius $\overline{BM}$ (see Sect. 2.6) :

$$\frac{\delta \overline{\mathrm{BM}}}{\overline{\mathrm{BM}}} = 3\frac{\delta B}{B} - \frac{\delta T}{T} \tag{2.53}$$

For fixed ∇ and T, the approximation formula may be simplified:

$$\frac{\delta\overline{\mathrm{BM}}}{\overline{\mathrm{BM}}} = 3\frac{\delta B}{B} \tag{2.54}$$

and assuming $\overline{KG}$ unchanged ($\delta\overline{KG} = 0$) the following important formula is derived:

$$\delta(\overline{GM}) = \delta(\overline{BM}) = \overline{BM}\cdot 3\frac{\delta B}{B} \tag{2.55}$$

Therefore, an *increase of beam* by 10 % leads approximately to an *increase of* $\overline{GM}$ by 30 %.

Finally, for the selection of the side depth D the key point is to achieve the required hold volume of the ship and to satisfy the Load Line regulations, namely, to reach the required minimum freeboard. Other influential factors are as follows:

- An increase of the side depth D involves an increase of the ship's gravity center $\overline{KG}$ and consequently a reduction of $\overline{GM}$ (negative influence on the *initial* stability). However, as to the *large angle stability*, we have an increase of the *range of the righting lever* due to the delayed immersion of the side deck and of the superstructures in water.
- An increase of D involves an increase in the modulus of the midship section. Therefore, for fixed L, due to the reduction of the occurring bending stresses on the ship's extremes (deck and bottom), there is a possibility to reduce the thickness of the plating and hence of the weight of the steel structure (see Sect. 2.7).

The L/D ratio can be selected from similar ships or in accordance with typical values of Table 2.7.

2.6 Selection of Beam

As has been pointed out earlier, the proper procedure of selecting the ship's main dimensions and of other fundamental ship values is to proceed, after the determination of the length, with the selection of the block coefficient C_B and thereafter of the beam, together with the draft. The selection of the C_B coefficient will be elaborated later in Sect. 2.10.

Assuming that the length L and the block coefficient C_B are known (predetermined), as we may assume this also for the ship's displacement Δ in first approximation (and for the corresponding displaced volume ∇), then the following relationship holds for the product $B\cdot T$:

$$B\cdot T = \frac{\nabla}{L\cdot C_B}$$

that is the selection of beam B can be accomplished on the basis of the product $B \cdot T$. Thereby changes of the beam require inversely proportional changes in the draft and indirectly of the side depth D (due to the minimum freeboard requirements).

Alternatively, the beam selection can be done through the L/B ratio, either by using data of similar ships (see Table 2.7), as explained previously, or by using some relationships, which are presented below and are deduced from the analysis of data of ships built in the 1990s. These relationships provide the L/B ratio as a function of length L (m) (Friis et al. 2002).

For cargo ships with $50 \leq L \leq 200$ m:

$$L/B = 4 + 0.015 \cdot (L + 17) \tag{2.56}$$

For reefer ships with $60 \leq L \leq 180$ m:

$$L/B = 4 + 0.014 \cdot (L + 11) \tag{2.57}$$

For containerships with $100 \leq L \leq 200$ m:

$$L/B = 4 + 0.009 \cdot (L + 42) \tag{2.58}$$

For containerships with $L > 200$ m:

$$6.5 \leq L \leq 7.1$$

For bulk cargo carriers with $L \geq 120$ m:

$$L/B = 6$$

For tankers:

$$L/B = 5.5$$

For LPG and LNG ships with $L \geq 100$ m:

$$L/B = 5.7 + 0.002 \cdot (L - 100) \tag{2.59}$$

For Ro–Ro cargo ships with $L \geq 80$ m:

$$L/B = 5.5 + 0.0036 \cdot (L - 41) \tag{2.60}$$

For ROPAX ships with $L \geq 80$ m:

$$L/B = 5.5 + 0.0033 \cdot (L - 141) \tag{2.61}$$

Similar set of data and relationships for various types of ships are also listed in Appendix A.

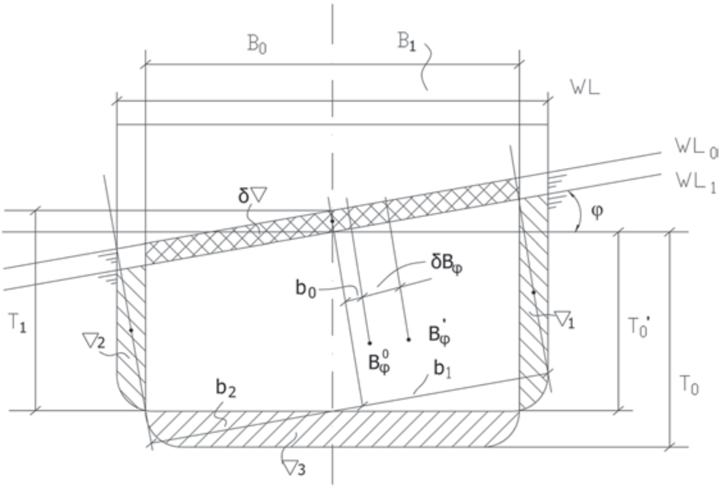


Fig. 2.34 Effect of change of beam on transverse stability

2.6.1 Effect of Beam on the Ship's Stability

To examine the influence of a change of the beam on stability, it is considered that the increase of beam is accompanied by a corresponding reduction of the draft, so that the displacement remains unchanged (see Fig. 2.34).

The ship is examined as inclined at an angle φ and with an initial waterline $W\mathrm{L}_0$ and center of buoyancy B_φ^o. At first, an increase of the beam implies an increase of the displaced volume by $(\nabla_1+\nabla_2)$. However, it is considered that the ship's draft decreases accordingly, namely becoming T'_0, so that the corresponding lost displaced volume ∇_3 balances the above increase $(\nabla_3=\nabla_1+\nabla_2)$.

The increase of the beam involves though an increase of the ship's steel weight by W_4, if we request an unchanged level of the ship's strength with respect to a maximum level of stresses on the ship's structure (see Sect. 2.6.3). This results in a new, weight increasing change of the ship's draft to the level of T_1 and the difference between the new and initial displacement (before the beam increase) is:

$$\delta W = W_1 + W_2 - W_4 \tag{2.62}$$

where

$W_1=w\cdot\nabla_1$, $W_2=\mathrm{w}\cdot\nabla_2$, w: specific water density
W_4: increase of steel weight due to increase of beam

Considering b_0, b_1, and b_2, namely the distances of the exerting centers of buoyancy of volumes $\delta\nabla$, ∇_1 and ∇_2 from the vertical line, which passes through the original center of buoyancy, we find for the shift of B_φ^o:

$$\delta B_{\varphi} = \frac{W_1 \cdot b_1 - W_2 \cdot b_2 + \delta W \cdot b_0}{W_0 + W_4} \tag{2.63}$$

where

$W_0 = w\nabla_0$: initial displacement,
$W_0 + W_4$: new displacement.

If it is assumed that there is no change of displacement, that is there is no weight increase W_4 from the beam increase, because, for example, of a possible simultaneous reduction of the ship's side depth, it is concluded:

$$\delta W = W_1 + W_2$$

and

$$\delta B_{\varphi} = \frac{W_1 \cdot b_1 - W_2 \cdot b_2 + \delta W \cdot b_0}{W_0}$$

The influence of an increase of the beam on the ship's initial stability, that is the $\overline{GM}$, can be analyzed as follows.

We consider that the ratio of change of beam $\beta = B_1/B_0$ is given; furthermore, the displacement and the other main dimensions L and T remain fixed. Then, the vertical prismatic coefficient C_{PV} remains unchanged:

$$(C_{PV})_1 = \frac{\nabla_1}{A_{WP1} \cdot T_1} = \frac{(C_B)_1}{(C_{WP})_1} = (C_{PV})_0 = \frac{(C_B)_0}{(C_{WP})_0}$$

Thus the block coefficient due to the beam increase is concluded:

$$(C_B)_1 = \frac{\nabla_1}{L_1 B_1 T_1} = \frac{\nabla_0}{\beta L_0 B_0 T_0} = \frac{(C_B)_0}{\beta} \tag{2.64}$$

and accordingly the water plane area coefficient:

$$(C_{WP})_1 = (C_{WP})_0 \cdot \beta^{-1}.$$

Recalling the well-known relation:

$$\overline{GM} = \overline{KB} + \overline{BM} - \overline{KG} \tag{2.65}$$

where according to Morrish

$$\overline{KB} \cong T(2.5 - C_{PV})/3 \tag{2.66}$$

or

$$\cong T-(1/3)\cdot\{(\mathrm{T}/2)+(\nabla/A_{\mathrm{WP}})\} \tag{2.67}$$

and

$$\overline{KG}=k_{\mathrm{D}}\cdot D, \tag{2.68}$$

where k_{D}: coefficient obtained from similar ships (see Table 2.15, Sect. 2.10.6); it is noted that neither $\overline{KB}$ nor $\overline{KG}$ are directly dependent on the beam B. Thus, looking into the analysis of $\overline{BM}$:

$$\overline{BM}=I_{\mathrm{T}}/\nabla \tag{2.69}$$

We assume for the moment of inertia of the water plane area about the longitudinal axis:

$$I_{\mathrm{T}}\cong k_{T}\cdot L\cdot B^{3} \tag{2.70}$$

where k_{T}: form coefficient of specific water plane $\cong 0.04 \div 0.06$ for ordinary water plane of shiplike forms.

For constant values of ∇, it may be assumed that for small changes of B, the coefficient k_{T} remains unchanged, thus:

$$(I_{\mathrm{T}})_{1}=k_{\mathrm{T}}\cdot L\cdot B^{3}{}_{1}=k_{\mathrm{T}}\cdot L\cdot\beta^{3}\mathrm{B}^{3}=(I_{\mathrm{T}})_{0}\cdot\beta^{3}$$

If we set:

$$(\overline{BM})_{1}=(\overline{BM})_{0}+\delta(\overline{BM})=(I_{T})_{1}/\nabla=\beta^{3}(\overline{BM})_{0}$$

and

$$\beta=\left(B_{0}+\delta B\right)/B_{0}=1+\delta B/B$$
$$\beta^{3}\cong 1+3\delta B/B+\ldots$$

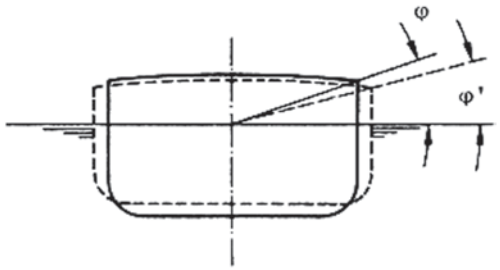
it is concluded that:

$$\delta(\overline{\mathrm{BM}})/\overline{\mathrm{BM}}=3\cdot\delta B/B$$

and if the vertical distribution of weights is assumed unchanged, that is $\delta(\overline{KG})=0$, we obtain consequently:

$$\delta(\overline{GM})=\delta(\overline{BM})=\overline{BM}\cdot 3\delta B/B$$

Fig. 2.35 Effect of increasing the beam on stability for constant midship section area—premature immersion of deck edge



Therefore, it is concluded, from the above hypotheses that an increase of beam by 5 % leads approximately to an increase of $\overline{GM}$ by about 15 %.

In the above reasoning the draft was considered fixed, but we had a change of C_B by the ratio $1/\beta$. If on the contrary the draft changes by the ratio $(1/\beta)$ and C_B remains fixed, like the displacement, then, with the increase of beam we have a small decrease of $\overline{KB}$ (due to the reduction of T), a drastic increase of $\overline{BM}$, as above, and finally a relative reduction in $\overline{KG}$, all this leading again to a significant increase of $\overline{GM}$.

Regarding the stability at large angles, if in parallel to the beam increase the draft decreases accordingly, and consequently the side depth, so that the displacement remains constant, the edge of the side deck apparently will immerse in water at smaller angles. However, for fixed midship section area but increased *B/D* ratio, this results in general in an increase of the range of stability as well as in a larger peak value (GZmax) of the righting lever, so as to compensate for the negative effect of the premature immersion of the side deck (Figs. 2.35 and 2.36).

2.6.2 *Effect of Beam on the Ship's Resistance*

Generally it may be expected that an increase of the ship's beam or the *B/T* ratio leads to higher resistance (primarily due to the increase of wave resistance) and

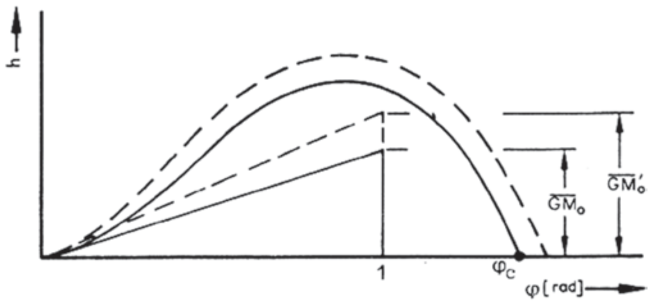


Fig. 2.36 Effect of increasing the beam on righting/restoring arm $h = GZ(\varphi)$: increase of value of initial stability $(\overline{GM})$ and usually increase of *GZmax* and of the range of the stability (increase of angle of vanishing stability φ_C)

hence of the required propulsion power. But such considerations are very general and prove often not true, if other ship dimensional parameters, in addition to *B/T*, are not taken into account in parallel.

For fast ships with a large proportion of residuary resistance, it has been shown that an increased beam generates more intense free surface disturbances and waves, thus higher wave resistance; this is due to larger inclinations of the waterlines with respect to the ship's symmetry plan and direction of advance. On the contrary, for slow ships (small Froude number) with relatively high frictional resistance percentage, it is recommended (for given displaced volume) to target an *as small as possible* wetted surface, which implies a ratio (*B/T*) corresponding to approximately 2.5 and a block coefficient of $C_B \cong 0.80$ (noting the block coefficient of a floating semisphere $= \pi/4$)[10]. However, it is considered that for fast ships (larger Froude number) the total resistance is also minimized for $B/T \cong 2.5$.

From the research of *Mumford and Moor* it was shown for the dependence of the ship's total resistance on changes of beam and draft (see Papanikolaou 2009a):

$$\frac{(R_T)_1}{(R_T)_0} = \left(\frac{B_1}{B_0}\right)^x \cdot \left(\frac{T_1}{T_0}\right)^y \tag{2.71}$$

where the semiempirical exponents x and y are given in the above table as a function of the Froude number and ship type (Table 2.8). The below semiempirical coefficients of Mumford were recently revised for Ro-Ro cargo and Ro-Ro passenger ships (Alissafaki 2013) (below Table 2.9).

Let us now have a look at the following seemingly hydrodynamic "paradox" that should defy the general impression of a negative effect of low *L/B* on the ship's resistance. If in parallel to the increase of the ship's beam the hull form changes in such a way that the design draft remains constant (and the same is assumed for the displacement and the length) then despite the reduction of the *L/B* ratio and the increase of *B/T* ratio a *decrease* of C_B (and C_P) and often a *reduction* of the residuary resistance is obtained. The following example of a reefer cargo ship, for which the

Table 2.8 Exponents by Mumford $x = f(F_n)$, $y = f(F_n)$ for cargo ships and fishing vessels

F_n		≤0.24	0.25	0.26	0.27	0.28	0.29
Cargo ship	x	0.85	0.85	0.85	0.85	0.85	0.85
	y	0.52	0.53	0.55	0.64	0.74	0.78
Fishing vessel	x	0.74	0.74	0.74	0.74	0.74	0.745
	y	0.60	0.60	0.60	0.60	0.60	0.60
F_n		0.30	0.31	0.32	0.33	0.34	0.35
Cargo ship	x	0.855	0.880	0.945	1.00	–	–
	y	0.80	0.80	0.80	0.80	–	–
Fishing vessel	x	0.78	0.84	0.90	0.95	0.995	1.00
	y	0.60	0.60	0.60	0.60	0.60	0.61

[10] It is can be readily shown that the (semi)sphere is the solid with the minimum surface area for a given enclosed volume.

Table 2.9 Exponents by Mumford $x=f(F_n)$, $y=f(F_n)$ for Ro-Ro cargo and Ro-Ro passenger ships

F_n		0.18	0.19	0.20	0.21	0.22	0.23
Ro-Ro ship	x	0.743	0.743	0.747	0.755	0.770	0.794
	y	0.368	0.371	0.377	0.384	0.395	0.408
Ro-Ro passenger ship	x	0.807	0.801	0.796	0.794	0.797	0.802
	y	0.307	0.309	0.313	0.317	0.324	0.333
F_n		0.24	0.25	0.26	0.27	0.28	0.29
Ro-Ro ship	x	0.818	0.835	0.854	0.893	0.963	1.053
	y	0.423	0.440	0.461	0.486	0.513	0.540
Ro-Ro passenger ship	x	0.807	0.815	0.835	0.874	0.925	0.975
	y	0.344	0.358	0.374	0.393	0.413	0.432
F_n		0.30	0.31	0.32	0.33	0.34	0.35
Ro-Ro ship	x	1.140	1.202	1.233	1.238	1.229	1.217
	y	0.564	0.583	0.600	0.616	0.632	0.652
Ro-Ro passenger ship	x	1.012	1.032	1.039	1.039	1.040	1.046
	y	0.450	0.467	0.484	0.503	0.524	0.550

resistance has been calculated according to the well-known *Taylor–Gertler* semiempirical method, verifies the above consideration (Tables 2.10, 2.11, and 2.12).

Table 2.10 Variation of B and C_P times L: fixed

L/B	6.5	7.0	7.5
B/T	3.2	3.0	2.8
C_P	0.56	0.60	0.65
$\frac{(R_T)_1}{(R_T)_0}$	0.80	1.00	1.32

Table 2.11 Variation of L and C_P times B: fixed

L	124	132	140
L/B	6.6	7.0	7.4
$L/\nabla_{1/3}$	5.9	6.3	6.7
C_P	0.64	0.60	0.57
$\frac{(R_T)_1}{(R_T)_0}$	1.42	1.00	0.79

Table 2.12 Variation of L and B times C_P: fixed

L	124	132	140
B	20.1	18.9	17.8
L/B	6.2	7.0	7.85
B/T	3.2	3.0	2.8
$\frac{(R_T)_1}{(R_T)_0}$	1.18	1.00	0.93

Given Initial Data

Δ =9,480 t	∇ =9,200 m^3
L_{PP}=132 m	$L/\nabla^{1/3}$=6.3
B =18.9 m	L/B =7.0
T =6.3 m	B/T =3.0
C_P =0.603	C_B =0.585
C_M=0.970	
V =22 kn	F_n =0.315

Parametric changes It is considered that the draft T and the coefficient C_M remain fixed, hence:

$$(\nabla / T \cdot C_M) = L \cdot B \cdot C_P \text{ : fixed}$$

1. Variation of B and C_P times L: fixed (Table 2.10)
2. Variation of L and C_P times B: fixed (Table 2.11)
3. Variation of L and B times C_P: fixed

Conclusions (by inspection of results of Tables 2.10, 2.11, 2.12)

1. Increase of the length always positively affects the resistance (reduction).
2. Reduction of C_P implies also reduction of the resistance.
3. *For given length (and draft), increase of beam, but also reduction of C_P, means reduction of the resistance* (see values in the above Table 2.10).
4. The influence of B/T on resistance (see above Tables 2.10 and 2.11) is herein almost negligible.
5. Note that the above conclusions cannot be generalized for other case scenarios, especially for ships designed for different Froude numbers, e.g. for tankers.

2.6.3 *Effect of Beam on the Ship's Structural Weight*

To investigate the beam's influence on the steel weight we assume at first that the moment of inertia of the midship section can be expressed approximately by the formula[11] (see Sect. 2.3.2):

$$\mathrm{I} = k \cdot A_f \cdot d^2 = k \cdot p \cdot t \cdot d^2$$

where

d: distance of the midship section's extremes from the neutral axis

[11] Assuming ship's structure represented by a bending beam and her midship section approximated by the cross section of an equivalent tubular beam, of mean thickness t and perimeter p.

A_f: cross area of the structure at the midship section
p: cross-section's perimeter
t: average thickness
k: form coefficient accounting for the form of the midship section

Assuming that the bending moment and the level of stresses on the midship section remain unchanged, which results from the requirement of fixed length, and additionally that the distribution of the structural elements of the ship's structure and the form of the midship section do also not change significantly, which means that the distance d and the form coefficient k remain constant, the following is concluded:

$$p \cdot t: \textit{ remains unchanged}.$$

Thus if we can set approximately for the perimeter:

$$p \cong 2(\mathrm{B} + D)$$

or for the under study ship:

$$p_1 = 2(\mathrm{B}_1 + D_1)$$

and in particular if the side depth remains unchanged, that is it does not decrease inversely with beam's increase:

$$p_1 = 2(\mathrm{B}_1 + D) = 2(\mathrm{B} + \delta\mathrm{B} + D) \tag{2.72}$$

it is concluded from the requirement:

$$(p \cdot t)_1 = (p \cdot t)_0$$

due to

$$\left(\frac{p_1}{p_0}\right) = 1 + \frac{\delta\mathrm{B}}{(\mathrm{B} + D)}$$

for the average thickness:

$$\left(\frac{t_1}{t_0}\right) = \left(1 + \frac{\delta B}{(B + D)}\right)^{-1} \cong 1 - \frac{\delta B}{(B + D)} + \ldots$$

Thus, if the steel weight of the main hull is approximated with the formulas (see Sect. 2.3.2)

$$W_\mathrm{H} = k_H \cdot A_\mathrm{H} \cdot t,$$

where the area of the ship's hull is assumed according to Taylor as following:

$$A_{\mathrm{H}} = C_{\mathrm{H}} \cdot \sqrt{\nabla \cdot L}$$

with

$$C_{\mathrm{H}} = f(\mathrm{B}/D, C_{\mathrm{MD}})$$

it is concluded for the weight:

$$(W_H)_1 = k_{\mathrm{H}} \sqrt{\nabla \cdot L} \cdot (C_{\mathrm{H}})_1 \cdot t_1 \quad (2.73)$$

From the last relationship it is shown that:

- The hull surface area increases with the increase of the ratio *B/D*, as shown by the dependence of the C_{H} coefficient.
- With the increase of beam by δB the average thickness of the plates decreases, provided that the side depth *D* and the ratio *L/D* are constant.
- The above two changes act in a counterbalancing way with respect to the steel weight, resulting usually in a slight weight increase due to the more drastic increase of the hull surface area.
- Provided that the side depth does not remain constant, but decreases inversely proportional with the increase of the beam, an increase of the plate thickness results, due to the increase of the ratio *L/D*, and of course an increase of the steel weight follows.
- Finally, an increase of beam around the midship section generally results in significant changes in the distribution of loadings due to higher concentration of weight and hydrostatic forces at this ship position. Thus, an increase of the bending moment at the midship section is concluded, resulting in a requirement for additional increase of the average thickness of the plates so as to achieve the required modulus and keep the maximum level of stresses unchanged; in view of the above, an increase of the steel weight and hence of the cost of the ship is concluded.

2.6.4 Other Factors Affecting the Selection of the Beam

1. **Behavior in waves:** When determining the ship's beam based on initial stability criteria, that is aiming at a satisfactory $\overline{GM}$, the behavior of the ship in waves and in particular the roll motions must be taken into account. For safety and operability reasons, such as avoidance of passengers' and crew's nausea, wave induced loadings on the ship's structure, equipment and cargo, speed loss due to excessive motions and added resistance in waves, problems of dynamic stability, and possible capsizing, we should be aiming at:

- Reduced roll motion amplitudes
- Reduced accelerations due to roll motion, especially in the transverse direction at larger distance from the vessel's rolling axis (e.g., deck area, where containers may be stowed)

The roll motion period of the ship depends at first on the period of the incident sea waves, exciting the ship's motions. However, if we restrict ourselves to the consideration of the most critical situation of resonance/tuning of the incident wave period with the natural rolling period of the ship, where the motions and accelerations are maximized, we may consider the natural rolling period of the ship

$$T_\varphi = \frac{2\pi i_\varphi}{\sqrt{g\overline{GM}}} \tag{2.74}$$

where

T_φ (second): natural period of rolling,

i_φ (meter): radius of the mass moment of inertia of the ship including the hydrodynamic, added mass moment, about the rolling axis

$$i_\varphi = k_\varphi \cdot B$$

where

k_φ $=0.32$–0.45[12], depending on the type and size of the ship,
i_φ $\cong 0.38B$ (average value)
g (m/s^2): acceleration of gravity
$\overline{\mathrm{GM}}$ (m): metacentric height.

If we assume that: $\overline{GM} \propto B^3$ and $i_\varphi \propto B$, it is concluded that:

$$T_\varphi \propto \frac{\mathrm{B}}{\sqrt{\overline{GM}}} \propto \frac{1}{\sqrt{B}}$$

that is, relatively large beam and high $\overline{GM}$ lead to small natural period of roll, which may be tuned with low wave periods, corresponding to short-length waves (Lewis 1988).

Low rolling periods induce *high* transverse accelerations, especially at points far away from the rolling axis (higher up on the ship's deck/superstructure). The ship's rolling axis is not fixed, but changes continuously its position in between the ship's still water plane and the vertical position of the center of mass of the ship. Indicative

[12] Low values hold for ships with their mass (ship's light ship mass plus deadweight) being concentrated in the holds region, for example bulkcarriers, particularly ore carriers; high values that may exceed even 0.45 hold for ships with voluminous and very high up extended superstructures, for example large cruise ships and to a certain extent RoPax ships.

Table 2.13 Typical natural roll periods for various types of merchant ships

Cargo ships	12–8 s
Coastal cargo ships	7–10 s
Bulk carriers	12–20 s
Tankers	~20 s
Reefer ships	16–18 s
Cruise ships	~20 s
RoPax ferries	10–14 s
Trawler/fishing vessels	10–13 s
Open sea tugboats	8–12 s

values for the natural period of roll motions of common types of commercial ships are listed in Table 2.13.

Finally, *Kempf* recommended the use of the so-called “roll number” (German: “Rollzahl”), which is defined as:

$$R = \frac{2\pi \cdot i_{\varphi}}{\sqrt{B \cdot \overline{GM}}} \tag{2.75}$$

which is a nondimensional value and varies between 8 and 14 for ships with good performance in waves. It is observed that for modern RoPax ships the value of R is often smaller than 8 due to the stringent requirements of intact stability regulations (requiring high $\overline{GM}$).

The negative effect of large amplitude roll motions on the operability of ships, especially those transporting sensitive cargos, passengers, or of naval ships, can be mitigated significantly by installing antirolling devices, such as antirolling fins, antirolling tanks, or/and simply bilge keels.

2. **Restrictions of beam for certain ship types:**

- Ships transporting standardized and bulky cargos (break bulk and unitized cargo), such as Ro-Ro, Ro-Pax, containerships, LASH, rail ferries, etc., require beams corresponding to the specific number of units of stowed cargo in the transverse direction. For container ships there is some degree of flexibility because of the existence of the side tanks and the relatively small width of a standard container (8 ft).
- Restrictions may apply to ships that operate through specific canals or channels (e.g., PANAMAX and SUEZMAX ships) or are being serviced by specific slipways of yards.
- From the restriction of draft for certain ship types, such as large tankers, bulk carriers, reefer ships, river ships, an increase of beam, or of the B/T ratio may be concluded, which leads often to undesired high values.

3. **Maneuverability performance:**

- It is considered that in general the reduction of the L/B ratio (increased beam) leads to an improvement of maneuverability of the ship, particularly regard-

ing the turning ability within a small diameter circle, in contrast to the course stability, which becomes generally worse.

2.7 Selection of the Side Depth

The side depth *D* of the ship's main deck is crucial for two fundamental ship properties:

- The available holds' volume
- The achieved freeboard

It is obvious that the selection of the side depth is inherently linked to the permissible draft. Indirectly it is related to the ship's length, in consideration of the ship's longitudinal strength, and beam, in terms of the stability of the ship.

It is considered that the side depth is the "*cheapest*" and *less problematic* main dimension of a ship. In particular, increase of side depth by 10 % causes an increase of the steel weight by 8 % for $L/D = 10$ or by 4 % for $L/D = 14$ (Schneekluth 1985), that is, the achievable volume increases more rapidly than the resultant increase of the ship's structural weight; consequently it is appropriate to prefer an increase of the ship's side depth rather than changes of other main dimensions, in case the ship's hold volume is inadequate.

2.7.1 Effect of Safety Regulations on Side Depth

The selection of side depth is significantly influenced by the following regulations regarding safety and operation:

1. The International Load Line Convention (ICLL 1988) that determines the freeboard deck and the permissible freeboard, namely the allowable difference between side depth and draft.
2. Regulations regarding the watertight subdivision of ships (International Convention for the Safety Of Life at Sea—SOLAS), which determine the subdivision (or bulkhead) deck of the ship. This regulation mainly affects the selection of the side depth of passenger ships, but also of some types of cargo ships, such as tankers longer than 150 m (ship of type A according to the Load Line Convention) and other dry cargo ships (type B ships) with reductions of the required freeboard (B-60 and B-100 bulk carriers). Certainly, when deciding on the watertight subdivision of a ship based on the floodable lengths curve, a relatively high position of the bulkhead deck and fewer bulkheads should be preferred, rather than vice versa.
3. Regulations of tonnage measurement (National and International Regulations—tonnage mark) affect the position of the main deck less than in former times, due to the more rational method of determining the ship's enclosed–exploitable spaces regardless of the existence of "tonnage openings" (see older types of cargo ships with a "shelter" deck, Antoniou and Perras 1984).

4. Regulations of classification societies specify an upper limit for the *L/D* ratio, which usually ranges between 14 and 16. If the upper limit of $L/D = 14–16$ (depending on the classification society) is not observed, then a dedicated examination of longitudinal strength and approval by the classification society is required. Particularly for certain small coastal ships or barge and bulk carriers operating in sheltered areas (e.g., Great Lakes ships), *L/D* ratios up to 20 have been approved in the past by some classification societies (e.g., ABS).

2.7.2 Effect of Side Depth on Hold Volume and Arrangement

As stated before, an increase of the side depth involves an increase of the available hold volume or of the *capacity factor* (Räumte), which expresses the ratio of the available grain hold volume to the ship's deadweight. Thereby, while an increase of the ship's length involves in general the synchronous increase of the ship's displacement, the increase of the ship's side depth results in an expansion of the available volume vertically and has no significant direct influence on the ship's displacement, besides causing a small increase of the ship's steel weight, if all the other dimensions remain constant. The height of the ship's main hull is very important for cargo ships, which, depending on the type of carried cargo, may be horizontally subdivided by intermediate decks at different levels.

Typically we refer to Ro/Ro cargo ships, ferries, reefer ships, as well as to conventional general cargo ships, which, for easy stowage and unloading reasons, dispose intermediate decks through which the ship is subdivided in the vertical direction. Obviously, the number and the exploitable height of the intermediate decks are determined by the cargo type and stowage method. Thus, while for general cargo ships there is some flexibility as to the available height of decks, this is not the case for Ro/Ro ships, car/train ferries, and reefer ships, where the height is determined by the cargo's standard dimensions and stowage/loading–unloading method.

Finally, for bulky cargo units, with standard dimensions, there are specific requirements as to the height of the side depth. Typically, the side depth of ships carrying standard containers is determined by the number of vertically stackable containers (height of 8′ to 8.5′ per unit). Here, the height of the coamings of the hatchways as well as the height of double bottom are taken into account. With the same criterion in mind, modern multipurpose/semi-container ships dispose similar side deck heights, so as to enable them to transport an integer number of containers in the area of the openings of their hatchways.

2.7.3 Effect of Side Depth on the Ship's Stability

The influence of the side depth on the ship's stability is complex and should be examined separately for the initial stability and stability at large angles.

With side depth's increase, the steel weight of the structure above main deck increases, resulting in raising the corresponding center of gravity. Also, the weight centers of superstructures and outfitting increase accordingly, leading to an increase of the ship's total $\overline{KG}$ in both light ship and fully loaded conditions. Thus, for small inclination angles, an increase of the side depth generally reduces the values of restoring moment (reduction of $\overline{GM}$) until the angle corresponding to the immersion of the main deck's edge.

After this angle, however, a significant increase of the stability righting arm is achieved, as well as an expansion of the region of positive values of restoring moment (range of stability), compared to the original ship.

Special attention should be paid to the selection of the side depth of RoPax ships, given that this value determines the main car deck, up to which the ship is considered vertically watertight. After the tragic accident of the RoPax ship *Estonia* (1994) very strict regulations on damage stability were established, explicitly taking into account the effect of possible water flooding on car deck due to sea wave impact in case the outer shell of RoPax ships is damaged (the so-called Stockholm Agreement, Papanikolaou 2002). The amount of water assumed flooding the car deck is a function of both the significant wave height in the area of ship operation and her freeboard in damaged condition[13]. Therefore, the selection of the ship's side depth (and her freeboard along with the selection of draft) is the result of a combined study of Load Line Regulations and damage stability requirements.

In conclusion, an increase of the ship's side depth adversely affects the stability at small inclination angles, whereas for large angles it has a positive effect when accompanied by sufficient freeboard. Generally, the magnitude of the side depth is determined by the amount and stowage of the transported cargo; possible stability problems in the course of ship design must be treated with other, more drastic means, for example adjustment of the ship's beam.

The selection of the freeboard, and thus of the difference between side depth and loaded draft, is addressed later in more details in Sect. 2.19.

2.7.4 Effect of Side Depth on the Ship's Structural Weight

If we assume the ship to be a bending girder (beam) (see Sect. 2.3.2) and examine its longitudinal strength, it is clear that with the increase of the side depth D and reduction of the ratio L/D, a reduction of the bending stresses in general occurs due to an increase of the girder's modulus, while the bending moment remains constant for fixed length.

The sectional modulus of the girder increases due to the shifting of the masses of the deck and the ship's bottom away from the neutral axis. Thus, the thickness of

[13] Maximum significant wave height, to be considered, is 4.0 m (for North Sea conditions); below 1.5 m sign. wave height, it is assumed that no water can flood the car deck of a RoPax ferry, if it complies with SOLAS 90 damage stability regulations.

the plates can be reduced, and the steel weight per cubic meter volume, decreases significantly as well as the corresponding construction cost.

As we have elaborated previously, if we set for the steel weight of the ship (see Sect. 2.3.2):

$$W_{\mathrm{H}} = k_{\mathrm{H}} \cdot A_{\mathrm{H}} \cdot t$$

where

A_{H}: hull shell surface area
t: average thickness of plates
k_{H}: form coefficient of specific midship section

and the hull shell surface area is approximated by:

$$A_{\mathrm{H}} \cong k_{\mathrm{H}} \cdot p \cdot L \cong k_{\mathrm{H}} \cdot 2(B + D) \cdot L$$

where p: perimeter of midship section, it is concluded for the weight:

$$W_{\mathrm{H}} = \alpha_{\mathrm{H}} \cdot t + b_{H} \cdot D \cdot t$$

that is, the weight increases with the thickness t and side depth D. However, given the moment of inertia of the midship section:

$$I \propto p \cdot t \cdot d^2 \propto p \cdot t \cdot D^2$$

and the modulus

$$W = I / d \propto p \cdot t \cdot d \propto p \cdot t \cdot D$$

it is concluded for the bending stresses at the midship section (see Sect. 1.1.2)

$$\sigma = \frac{M_{\max}}{W} = \frac{\Delta \cdot L}{C \cdot W}.$$

As Δ, L, and C are fixed, it is clear that

$$\sigma \propto \frac{1}{p \cdot t \cdot D} \propto \frac{1}{a \cdot t \cdot D + b \cdot t \cdot D^2}.$$

That is, if the level of the stresses is considered unchanged, the reduction of the thickness t is very significant and inversely proportional to side depth D leading to a reduction of ship hull's steel weight W_{H}. Nevertheless, this result is based on simplified assumptions and may slightly change in practice, but without changing the identified general trends.

2.8 Selection of the Draft

As mentioned earlier, during the selection of the beam, following the estimation of the displacement, length and block coefficient, the product $B \cdot T$ is considered known. Thus, the selection of the beam (see Sect. 2.6) involves indirectly the selection of draft, namely through the selection of typical values of the *B/T* ratio (see Table 2.6, Sect. 2.3), assuming that the product $B \cdot T$ is not known from other sources. Also, following the selection of side depth, the maximum permissible draft is determined by the required freeboard, which is calculated based on the length *L*, the side depth *D*, C_B and various other ship particulars. The main factors affecting the selection of draft are analyzed in the following.

2.8.1 Effect of Draft on Resistance and Propulsion

The draft of the ship appreciably affects the components of the total resistance, that is the frictional and wave resistance, of both slow and fast ships.

As indicated in Sect. 2.6.2 regarding the reduction of frictional resistance, which dominates the total resistance for relatively slow ships of small Froude number, it is required to achieve a minimum wetted surface area, which can be shown to be associated with values $B/T = 2.0–2.5$, depending on the C_B and the form of sections at the ship's both ends.

In addition, in order to minimize the wave resistance, we aim at shifting displacement away from the water plane downwards, which results in slender hulls.

It has been verified by experiments that a ratio *B/T* around 2.5 serves best not only frictional resistance but also wave resistance aspects.

From the propulsion point of view, one as large as possible draft is always sought aiming at the fitting of a large diameter propeller, with good efficiency in view of the resulting moderate loading on the propeller blades and the low turning of the propeller (low RPM). This general rule applies to all types of ships and especially to towing ships (tug boats and fishing vessels). It should be, however, taken into account that for not fully loaded or bow trim conditions, a large diameter propeller tends to emerge more frequently than a smaller one. Finally, on certain types of ships it is not possible to install a large diameter propeller because of the required high RPM of propeller and engine (small boats), or in case of multipropeller ships. Generally, for twin-propeller ships the ratio *B/T* is higher than the corresponding one for single-propeller ships (>2.6 vs. 2.1–2.5).

2.8.2 Effect of Draft on Stability

The influence of draft on the stability is not as obvious as that of the beam and side depth. From the relationship:

$$\overline{GM} = \overline{KB} + \overline{BM} - \overline{KG},$$

it can be at first concluded that an increase of the draft positively affects ship's initial stability, as it essentially implies an increase of $\overline{KB}$. If the increase of draft is coupled with a swift of displacement towards the design water plane (V-type sections), namely by increasing the fullness of the water plane and beam, the influence on the initial stability is drastic because of the synchronous increase of $\overline{BM}$, whereas the $\overline{KG}$ does not increase significantly.

However, it should be noted that beyond the design process, where the displacement is presumed fixed, if we examine the stability for various loading conditions, the change of $\overline{BM}$ should be considered through

$$\overline{BM} = \frac{I_T}{\nabla}$$

namely, for an *increased* draft the *increase* of I_T is usually less drastic than the increase of displacement ∇, so as to conclude to a *decrease* of $\overline{BM}$, hence of $\overline{GM}$ [14].

2.8.3 Influence of Draft on Seakeeping and Maneuverability

The influence of draft on the ship's seakeeping performance is particularly important for the light loaded condition, for example ballast condition.

In order to avoid intense slamming in the ballast condition, the minimum draft at the bow should be:

$$T_F \geq 0.02L \tag{2.76}$$

Furthermore, in order to avoid the emergence of the propeller (propeller racing), the minimum draft at the stern is recommended to be:

$$T_A \geq D_P + e + 0.4m \tag{2.77}$$

where

D_P: propeller diameter

e: distance of the lower tips of the propeller blades from the baseline, $\cong$0.1–0.2 m (for ships without rudder-post).

Finally, in order to achieve sufficient maneuvering capability, the product of $(L \cdot T)$, that is the longitudinal projection (lateral plan) of the wetted hull surface, should be proportional to the projected rudder area A_R:

$$\frac{L \cdot T}{A_R} = 30 \div 80 \tag{2.78}$$

[14] Considering also the parallel change of KG as a function of ship's draft, especially its significant increase when moving from the light to the full load condition, it is obvious that the stability of a ship in ballast condition is generally less problematic than in full load condition.

where the values on the right are determined by the ship type (lower limit values: best maneuvering ships, like tug boats; upper limit: fast passenger ships, tankers, etc.).

2.8.4 Influence of Draft on Strength

In view of the negative influence of large lengths on the longitudinal strength and on torsional stresses (see Sect. 2.3.), the trend in certain modern ship designs is evident to account for relatively large drafts (and beams). Of course, this leads to higher hydrostatic pressures at the bottom, the strengthening of which involves an increase of the ship's steel weight.

However, if the other dimensions remain unchanged (or may be even reduced), the latter effect is not significant, compared to similar increases of weight due to changes of other dimensions. It should be noted, however, that if the beam increases in parallel to draft, in view of the large projected areas at the ship's bottom, it is likely that problems of "transverse strength" arise, which may require additional strengthening and may result in increases of the steel weight. In this case, however, a parallel decrease of the ship's length may be expected, what counterbalances this likely steel structure weight increase.

2.8.5 Effect of Route Limits

The draft is the main dimension of every ship that is most affected by the restriction of depths of navigating routes. The permissible ship draft is determined by the governing depths in the calling ports, entrance ways to ports, channels, canals, estuaries, bays, and narrow sea straits, considering in addition the effect of natural-periodic (e.g., tidal effects) or irregular fluctuations of sea surface. Generally, an increase of draft is undesirable by ship operators because of introduced limitations of navigation.

Restrictions on draft automatically lead to increases of other dimensions, mainly of beam (see Table 2.7, Sect. 2.3, shallow draft tankers and bulk carriers).

Characteristic limits of well-known channels, canals, rivers (Figs. 2.37, 2.38, and 2.39):

- Panama Canal: $T<13$ m (under dredging, up to 15.2 m until 2014)
- Suez Canal (Egypt): $T<18$ m (1984)
- Northeast Sea Channel (North Europe): $T<9.5$ m
- Canal St. Lorenz (USA–Canada): $T<7.6$ m
- Estuary of La Plata River (South America): <8.2 m

Fig. 2.37 The Panama Canal

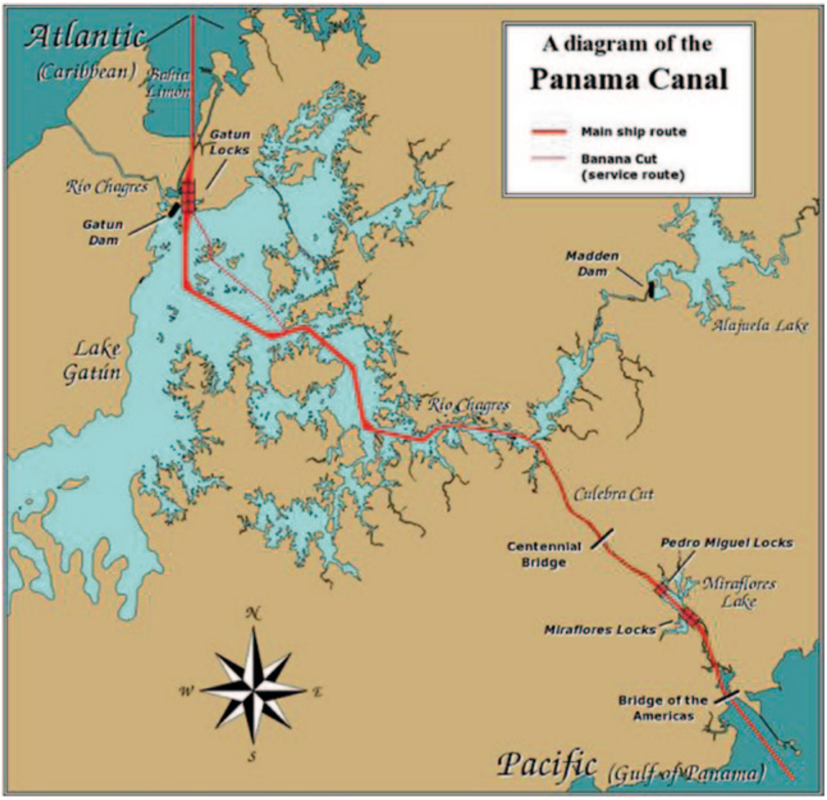


Fig. 2.38 Geography of the Panama Canal

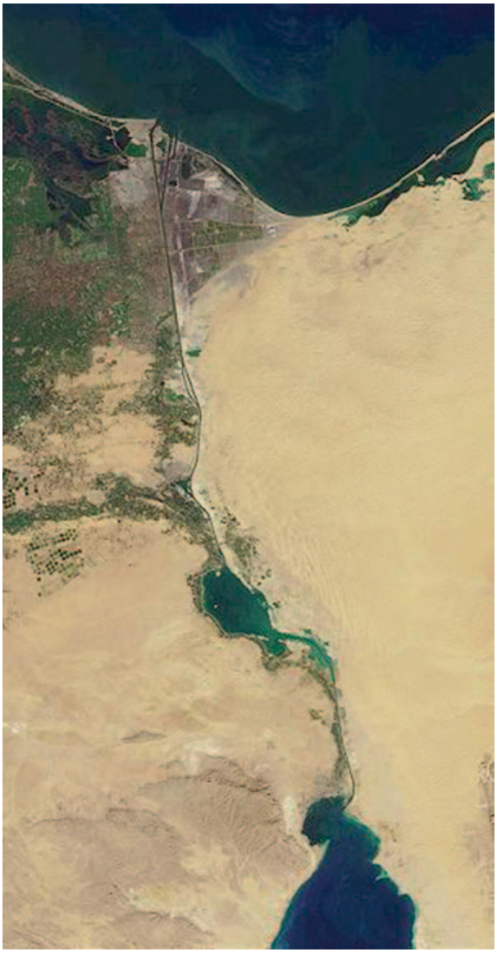


Fig. 2.39 Satellite photograph of the Suez Canal

2.9 Selection of Hull Form Coefficients

With the determination of the ship's block coefficient C_B, which generally expresses the fullness of the wetted part of the ship's volume compared to the volume of a rectangular parallelepiped of the same main dimensions L, B and T, the other hull form coefficients, such as the midship section coefficient C_M, the prismatic coefficient C_P, and finally the water plane coefficient C_{WP}, have been essentially also determined.

The coefficients affecting the selection of C_B also influence the selection of C_P, since both coefficients do not differ significantly, for common values of the midship section coefficient C_M varying between 0.94 and 0.99 for cargo and passenger ships (see Table 2.6); we may recall the well-known relationship

$$C_P = C_B / C_M$$

However, in a sense, the selection of C_P,

$$C_P = \nabla / (L \cdot A_M) \tag{2.79}$$

where A_M is the midship section area, should precede that of C_B, because C_P expresses more properly the fullness of the hull of the ship under study compared to that of a prismatic hull, of basis area A_M and height L. Particularly, small C_P means a concentration of displacement amidships and slender ends, whereas a large C_P corresponds to a relatively small midship section area, an even distribution of the displacement longitudinally and an extended parallel body amidships.

The midship section coefficient CM

$$C_M = A_M / (B \cdot T) \tag{2.80}$$

expresses the fullness of the midship section area in relation to the area of the circumscribed rectangle of the same B and T. Besides certain relatively small vessels with special requirements on stability and propulsion, namely need for sufficient draft for the installation of a propeller of as large as possible diameter, all other ships have very high C_M values (see Table 2.6). Small vessels that are exceptions from the above rule are fishing boats, tugboats, pilot boats, etc., with relatively small C_M (up to 0.70). For those vessels the difference between C_B and C_P is significant and attention should be paid during the preliminary design stage, when interpreting corresponding values.

Following empirical data of vessels *without* bottom deadrise, as it is common for large cargo ships, the following formulas, which correlate C_M with C_B, are recommended for use (Table 2.14):

For ships with a small L/B and bottom deadrise (such as fishing boats, tugs) the use of data from similar ships is recommended.

Finally as to the selection of the waterplane area coefficient C_{WP}, which influences the stability and wave-making resistance of the ship, both the fullness of the hull, namely C_B (or C_P) coefficient, and the form/character of the sections, also the bow type, should be taken into account. Generally the C_{WP} coefficient varies according to the variation of C_B (C_P).

The following formulas are concluded from empirical data:

a. **U-type sections**

$$C_{WP} = 0.778 C_B + 0.248 \tag{2.81}$$

$$C_{WP} = 0.95 C_P + 0.17(1 - C_P)^{1/3} \text{(Schneekluth)} \tag{2.82}$$

Table 2.14 Empirical data of vessels without bottom deadrise

V. Lammeren	$C_M = 0.9 + 0.1\ C_B$
H. Kerlen	$C_M = 1.006 - 0.0056\ C_B^{-3.56}$
HSVA Tank (Hamburg)	$C_M = 1/(1 + (1 - C_B)^{3.5})$

b. **Normal sections**

$$C_{\mathrm{WP}} = (1 + 2C_{\mathrm{B}})/3 \tag{2.83}$$

c. **V-type sections**

$$C_{\mathrm{WP}} = 0.743C_{\mathrm{B}} + 0.297 \tag{2.84}$$

$$C_{\mathrm{WP}} = (1 + 2C_{\mathrm{B}}/\sqrt{C_{\mathrm{M}}})/3 \text{ (Schneekluth)} \tag{2.85}$$

The above formulas are valid for cruiser stern ships, or ships with transom stern of limited extent. Newer constructions, with intense transom lines at waterline, have usually higher C_{WP} values, as can be seen from comparisons with similar ships. Typical values of the C_{WP} coefficient are presented in Table 2.6, Sect. 2.3.

2.10 Selection of Block Coefficient C_{B} and Prismatic Coefficient C_{P}

The *block coefficient* C_{B} (see Papanikolaou 2009a, Vol. 2 for all definitions) represents the ratio of the ship's displaced volume to the volume of the circumscribed rectangular parallelepiped with dimensions L (usually L_{PP}), B, and T. It can easily be shown that the C_{B} is the product of the *prismatic coefficient* C_{P} and *midship section* coefficient C_{M} (Fig. 2.40), i.e.,

$$C_{\mathrm{B}} = C_{\mathrm{P}} \cdot C_{\mathrm{M}}, \text{where } C_{\mathrm{B}} = \frac{\nabla}{LBT} \text{ and } C_{\mathrm{P}} = \frac{\nabla}{A_{\mathrm{M}}L}$$

Thus, if the midship section coefficient C_M does not change significantly, as typically happens to large and mainly bulky vessels, the C_P and C_B coefficients can be considered to be equivalent in terms of their meaning with respect to the slenderness of the hull form, exhibiting comparable values.

The prismatic coefficient C_{P} represents the ratio of the displaced volume to the volume of a prism with the basic area A_{M} (midship section area) and the height (=length) L (see also the following sketch; Fig. 2.41).

C_{P} describes the degree of concentration of the ship's displacement with respect to the midship section; however, the lengthwise distribution of the displacement cannot be concluded uniquely based on the value of C_{P} only. Nevertheless, small C_{P} generally indicates a ship with a relatively large area A_{M} and concentrated displacement around the midship section (thus slender ends), whereas large C_{P} means evenly distributed displacement along the ship length and long parallel body around the middle of the ship with short and bulky ends.

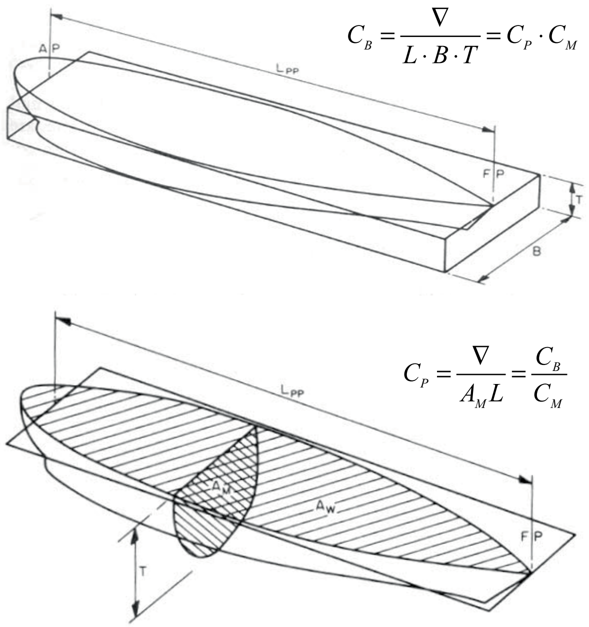


Fig. 2.40 Hull form coefficients C_B and C_P

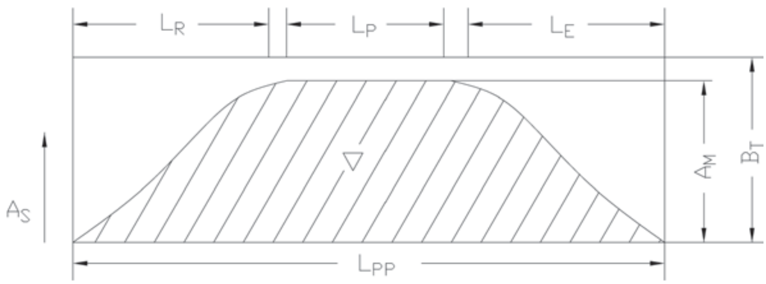


Fig. 2.41 Definition of sectional area curve

Particular attention is required when evaluating the true meaning of the information that the values of the coefficients C_B and C_P contain, especially when dealing with small vessels such as fishing boats, tugs, and speedboats. Here, the significant effect of the relatively small C_M must be assessed in parallel (see Fig. 2.42).

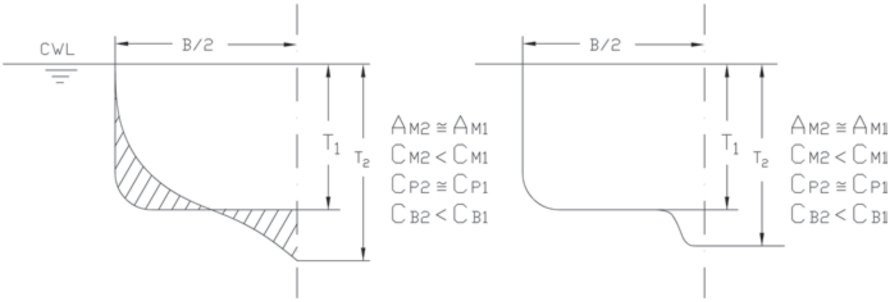


Fig. 2.42 Representativeness of block- and prismatic coefficients with respect to ship's hull form

Thus, in the above cases, while for the two hulls with $T_1 < T_2$ the prismatic coefficient remains, in both cases, almost unchanged (and the displacement also does not change), the block coefficients C_B differ significantly ($C_{B1} > C_{B2}$).

In conclusion, the prismatic coefficient describes more effectively the form of the hull and any review of the ship's hull geometry must take into account, in addition to $C_{B,}$ also the values of the coefficients C_P and C_M.

The slenderness ratio $L/\nabla^{1/3}$ complements the quantitative description of the wetted hull of the ship. The following examples demonstrate the importance of the coefficient C_P and ratio $L/\nabla^{1/3}$ in the assessment of the hull geometry of various types of ships:

a. Ocean liner—fast passenger ship: $L_{PP}/\nabla^{1/3} = 7.2$, $C_P = 0.57$
b. Fishing vessel—tugboat: $L_{PP}/\nabla^{1/3} = 5.2$, $C_P = 0.62$
c. River boat—cargo ship: $L_{PP}/\nabla^{1/3} = 6.8$, $C_P = 0.85$

From the above examples it is concluded that, only high values of slenderness ratios, accompanied by small C_P, lead to slender hulls.

2.10.1 *Effect of C_P and C_B on the Ship's Resistance*

The influence of C_P and C_B coefficients on the ship's resistance is significant. However, the factors affecting the selection of C_P (and C_B) differ depending on the corresponding operational Froude number.

For relatively slow ships (low Froude number), we try to minimize the wetted surface, as the objective is herein to keep the frictional resistance as low as possible, as in the total resistance breakdown this resistance component prevails significantly over the wave-making resistance. Thus, relatively high coefficients C_P (and C_B) and large midship sectional areas are concluded for tankers and bulkcarriers (C_P and C_B up to 0.88, C_M up to 0.99).

For relatively fast ships (high Froude number) it is necessary to reduce the more significant wave resistance as much as possible. The objective herein, is to control/tune the superposition of the various ship generated wave systems, especially those created at the ends (bow and stern) and the shoulders of the ship. The concentration of displacement in the middle of the ship generally leads to a smoothing of the

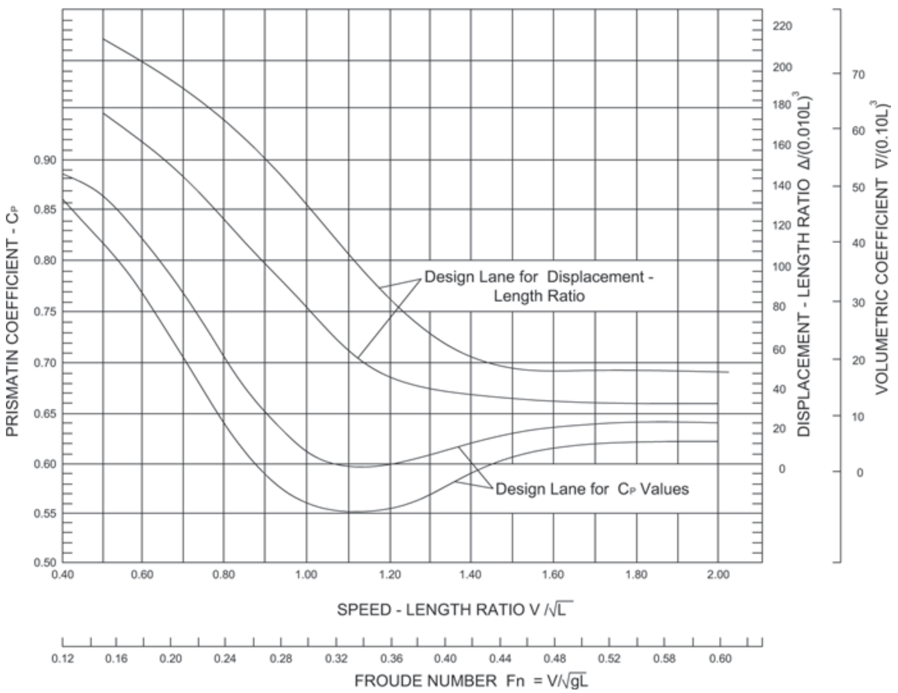


Fig. 2.43 Regions of variation of prismatic and volumetric coefficients for built ships by Saunders. (Lewis 1988)

shoulders and of the intensity of the corresponding secondary wave systems (see Sect. 2.3). For each length and displacement, thus for a given slenderness ratio, there is an optimal C_P as function of the Froude number, leading to a minimum wave resistance. Generally, for high Froude numbers, a low optimal C_P is concluded (see, for example, resistance curves of the systematic hull form series DTMB by Taylor–Gertler and FORMDATA by Guldhammer in Schneekluth 1985). However, if the Froude number exceeds a certain limit ($F_n \geq 0.33$), the total resistance only slightly varies with C_P, while for $F_n \geq 0.46$ it decreases slowly with *increasing* C_P. For ships with bulbous bow, the above limits may be shifted to higher values.

From the diagram below (Fig. 2.43), which shows the variation of C_P and of the volumetric coefficient versus the Froude number for built ships, the following is concluded:

1. **Slow ships ($F_n \leq 0.24$):** The prismatic coefficient is chosen to be relatively high, and in particular higher than the hydrodynamic optimum. Hence, the frictional resistance is minimized and the relatively low wave resistance does not increase significantly. Non-hydrodynamic aspects, such as construction cost and space exploitation, are positively affected by large C_P and dominate the selection of C_P.
2. **Fast ships ($0.24 \leq F_n \leq 0.36$):** In this region it is appropriate to choose C_P following hydrodynamic performance criteria, i.e., with a view of minimizing resistance. Thus, typical C_P values are actually close to the hydrodynamic optimal ones, given that the operational cost (greatly affected by powering and fuel consumption) can be shown equally important to the construction cost.

3. **High speed craft ($0.36 \leq F_n \leq 0.46$):** It can be seen that in this region the total resistance varies only slightly with C_P, thus the selection of C_P may be determined by other factors (including dynamic stability aspects).
4. **Speedboats and small crafts ($F_n > 0.46$):** For speedboats and small crafts the model experiments show that the total resistance decreases slightly with increasing C_P. Again other aspects determine the selection of C_P, with dynamic stability, lift and trim considerations now dominating.

2.10.2 Effect on the Seakeeping Performance

Besides low calm water resistance and propulsive power, the ultimate goal of a good ship hull designer is to achieve good performance in natural seaways, namely small ship motions (pitch, heave, roll, etc.) and accelerations (vertical and transversal), low added resistance and powering in waves, thus good seakeeping.

It is well known that large ship motions due to heavy seas, especially pitching and heaving, lead to added resistance and powering (added resistance can make up to 70% of the calm water resistance). Model experiments conducted by Todd (Schneekluth 1985) with a cruiser ship model of $C_B = 0.5$ travelling in head seas with constant propulsion power have shown that for an incident wave length $\lambda \cong 1.05\ L$, which corresponds approximately to the resonance region of heave/pitch motions, the *involuntary* loss of the ship's speed was 22% or $dV = 0.22\ V_0$; while for $C_B = 0.7$ (cargo ship) and the same displacement and length, the measured speed loss was 55% or $dV = 0.55\ V_0$. In addition to the above *involuntary* speed loss at constant propulsive power, dynamic loading on the steel structure, bow slamming, propeller racing, nausea of passengers, excessive loadings on the cargo, etc., may lead the ship's master to a *voluntary* reduction of the speed (decrease of propulsive power supply) to mitigate these phenomena.

The pitch motions of a ship take place about a time varying transverse axis, which passes near the center of floatation of the still waterplane and are the result of the forces and moments exerting on the vessel due to the changes of the hydrodynamic pressure distribution along the ship at her actual position with respect to the incident wave[15].

Generally, the influence of C_P on the amplitude of the resulting heave/pitch motions is not straightforward. The amplitude of the motions (and hence of accelerations) depends largely on the bow configuration/form (below and above waterplane), the length and the speed of the ship, as well as on the characteristics of the incident wave (height, heading and period/wavelength).

Apart from the unclear influence of a small C_P on heave and pitch motions, it has been shown that a small C_P tends to increase the probability of green-water. Thus, a relative increase of freeboard and of bow height is required, to counteract this

[15] The heave/pitch motions of a ship are strongly coupled to each other and generally have comparable values of natural period, which makes the tuning/resonance of both motions with the encountered wave very undesirable. This can be overcome only by changing the course and/or the speed of the ship.

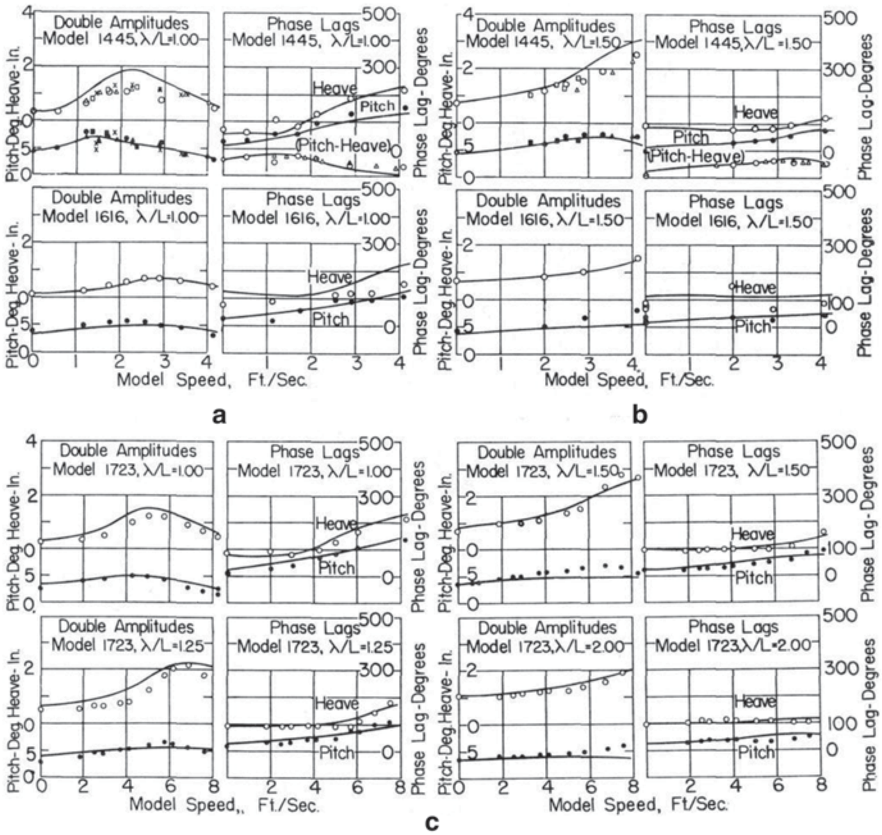


Fig. 2.44 *Double amplitudes* and *phase lags* of *pitch* and *heave* motions for two Series 60 models (**a**), (**b**) and one cruiser ship model (**c**) in head seas. (Lewis 1988). **Parameters:** Ratio of the incident wave length λ to the model length L and model speed; wave height for (**a**) and (**b**) is 1.25 in. and (**c**) 1.43 in. (model scale)

negative trend, as well as appropriate shaping (flare) of the bow sections above still water level. The study of the ship's seakeeping, thus also parametric studies regarding the effect of the ship's main design parameters on seakeeping, can nowadays be conducted by advanced numerical simulation methods and systematic model experiments (Fig. 2.44).

2.10.3 Effect on the Construction Cost

The construction effort to meet the requirements of a given hold volume (e.g., as determined by the transport capacity of the ship), increases for slender, sharply formed ships in terms of the weight of steel processed and the extent/weight of outfitting.

Generally, slender ships are characterized by larger steel areas per unit enclosed volume (especially for the outer hull shell), due to larger linear dimensions than bulky

and relatively short ships of the same displacement. Thus, regarding the construction effort and related costs, relatively high C_P and C_B coefficients should be favored.

2.10.4 Effect on the Exploitation of Spaces

The exploitation of hold's volume, especially with respect to the transport of standard/unitized and nonstandard break bulk cargoes (beyond the transport of containers that are transported in dedicated cellular type holds) significantly depends on the hull form of the ship and therefore on C_P and C_B.

The ideal hold space is bounded by large, unobstructed, and flat surfaces, both on the bottom and on the sides (vertical walls). Thus, small C_P and C_B coefficients, particularly in combination with V-type sections, seriously constraint the exploitation of spaces other than in the midship part.

Ships carrying standardized containers have the following peculiarity: whereas they carry a cargo that would best fit in boxlike holds and likewise hull form, their relatively high speed (and Froude number) calls for relatively small C_P and C_B coefficients; the practical solution to this problem is that at the ends of the ship the container cells are adjusted to the nonvertical side walls, so that losses of the exploitable volume are minimized to the extent possible (stepwise arrangement of containers, see below example of 3,400 TEU containership, Fig. 2.45).

Likewise, Ro-Ro ships and car ferries, with small C_P coefficients dispose reduced exploitation of the lower deck spaces in the bow region, because of their relatively high speed and sharp entrance of the waterlines in this region (see, Fig. 2.46).

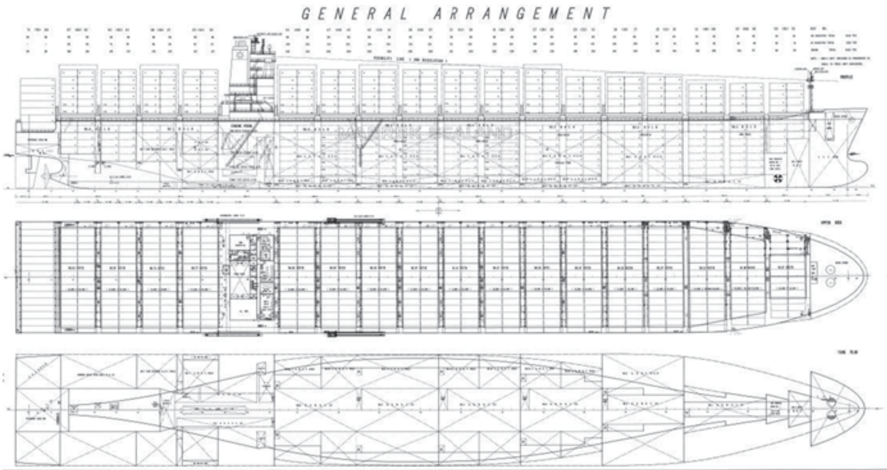


Fig. 2.45 General arrangement of 6300 TEU Containership (*Shipyard Hyundai Heavy Industries Co. Ltd.*)

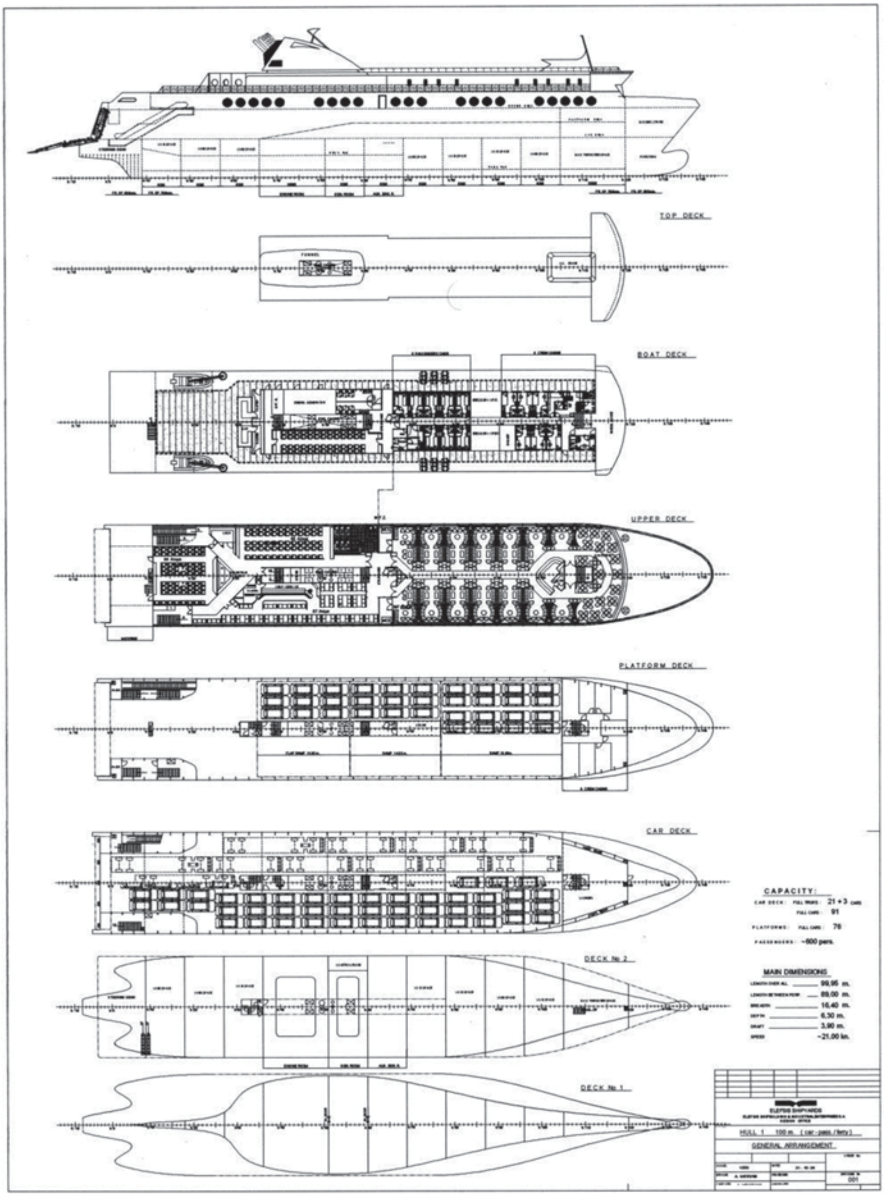


Fig. 2.46 General arrangement of a RoPax ship, Joint Industry-University RTD project EPAN–transport, NTUA-Elefsis Shipyard (2005–2007)

2.10.5 Effect on the Stability

The initial stability of the ship ($GM = KM - KG = KB + BM - KG$), the vertical position of the buoyancy center ($\overline{KB}$), and the moment of inertia of the waterplane

about the centerline (I_T, thus also *BM*) can be positively influenced by small C_P coefficients, which are combined with relatively large draft and beam as well as V-type sections.

Summary—Conclusions

Likewise in the selection of length, the basic factor affecting the determination of the C_B (and C_P) coefficient is the low resistance (and powering) of the ship, for the required speed, and in combination with the pre-estimated length, for the given Froude number. Generally: *high Froude number requires a low C_B (and C_P) coefficient for a hydrodynamically optimal ship.*

Other factors affecting the selection of C_B are: the weight and the cost of steel structure, the exploitation of cargo spaces, and the seakeeping behavior of the ship in waves (the motions and accelerations at various points of the ship, as well as the added resistance due to her motions in waves). In practice, like with the selection of the ship's length, C_B is selected differently from the optimal one with respect to least resistance, namely, usually larger values than those corresponding to hydrodynamically optimal solutions are preferred.

2.10.6 Approximate/Semiempirical Formulas

Common ways of estimating the value of C_B are:

A. Using semiempirical mathematical formulas from statistical data of built ships (considering both hydrodynamic and economic criteria).
B. Using semiempirical mathematical formulas from statistical analysis of ships of "minimum building cost for given deadweight (DWT) and speed."
C. Using diagrams based on mathematical formulas according to A or from statistical data of similar ships.

Notes

A. The employed semiempirical formulas have the following general form (in metric system):

$$C_B = K_1 - K_2 F_n - K_3 F_n^2 \tag{2.86}$$

where the coefficients K_1, K_2, K_3 are listed in Table 2.15 below (they may refer to the ship's *trial* speed V_T or *service* speed $V_S \approx 0.94\ V_T$).

Table 2.16 summarizes similar, well known formulas given in the Anglo-Saxon/British Imperial system (V [kn] and L [ft]), which take the general form:

$$C_B = K_4 - K_5 V/\sqrt{L} \tag{2.87}$$

where V is mainly the trial speed, unless otherwise noted.

Table 2.15 Coefficients of semiempirical formulas for the calculation of C_B (metric system units)

Formula	K_1	K_2	K_3	Comments
Horn	1.06	1.68	0	Single-screw ships, service speed
Ayre	1.08	1.68	0	Single-screw, trial speed
Ayre	1.09	1.68	0	Twin-screw, trial speed
Heckser	1.00	1.44	0	Single-screw, trial speed
V. Lammeren	1.08	1.68	0.224	Single-screw, trial speed

Table 2.16 Coefficients of semiempirical formulas for the estimation of C_B (Anglo-Saxon system of units)

Formula	K_4	K_5	Comments
Alexander and Watson	1.06	0.500	$0.65 \le V/\sqrt{L} \le 0.8$ (cargo ships)
	1.03	0.500	$V/\sqrt{L} > 0.89$ (fast cargo ships)
	1.12	0.500	$V/\sqrt{L} < 0.65$ (slow cargo ships)
Silverleaf and Dawson	1.214	0.394	bulky ships, $C_B \ge 0.75$, length L[m]
Chirila	1.225	0.378	bulky ships, $C_B \ge 0.75$, length L[m]
Troost	1.156	0.625	Service speed $V_S \cong 0.94\, V_T$

B. The below given formulas are derived from optimization studies of ships with respect to minimum building cost for given deadweight and speed (Schneekluth 1985):

$$C_B = \frac{0.14}{F_n} \cdot \frac{L/B + 20}{26} \tag{2.88}$$

$$C_B = \frac{0.23}{F_n^{2/3}} \cdot \frac{L/B + 20}{26} \tag{2.89}$$

The formulas are valid for $0.14 \le F_n \le 0.32$ and are limited to ships with $0.48 \le C_B \le 0.85$.

C. Finally, the following diagrams or comparable graphs of $C_B = f(F_n)$ as a function of the ship type (see Figs. 2.47 and 2.48) can also be used.

2.11 Midship Section Coefficient C_M

The midship section coefficient C_M, which, as mentioned above, connects the most important hull form coefficients C_B and C_P, can be selected quite freely by the designer, taking into account some basic factors such as low resistance, ease for construction, space exploitation, and sufficient stability.For a given midship section area A_M, B, and T, thus also fixed C_M, the possibility of alternative configuration of the midship section is associated with the selection of the bilge radius and the deadrise of the bottom (see Fig. 2.49).

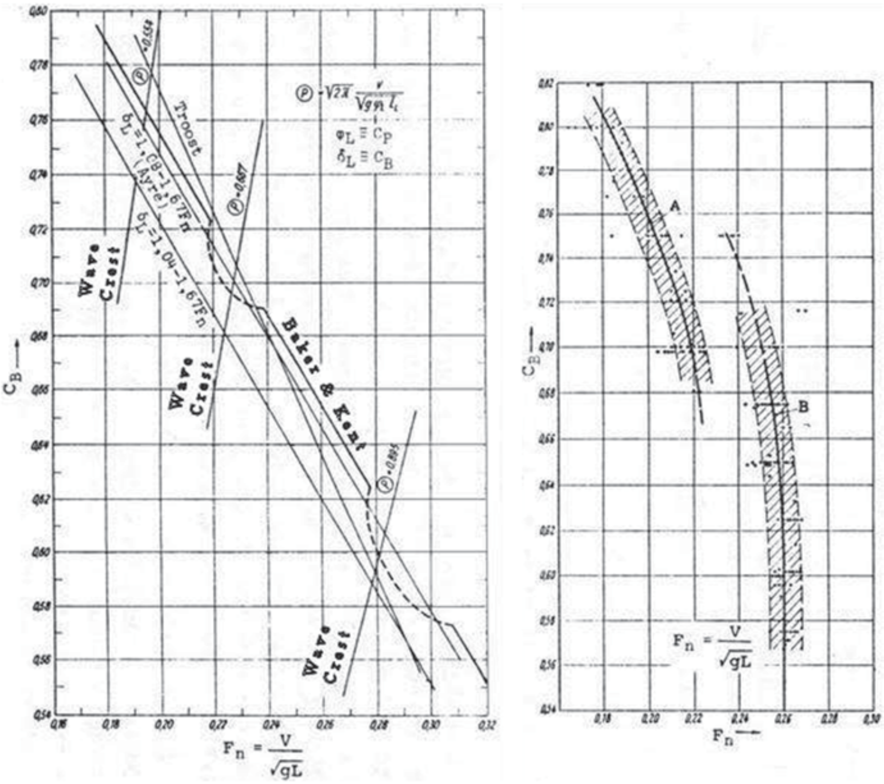


Fig. 2.47 Block coefficients C_B versus Froude number. (**a**) Regions for the favorable selection of C_B to avoid tuning of ship generated, bound waves according to Baker and Kent (**b**) Regions for the selection of C_B and statistical data according to Danckwardt for slow *(A)* and fast (*B*) ships

2.11.1 Effect on Resistance

The influence of C_M on the total resistance of a ship is considered to be small, but results indirectly through the C_P, for given displacement and main dimensions. The individual effects of C_M are:

a. For slow ships with substantial frictional resistance, a minimization of the wetted surface is targeted. Therefore, for unrestricted beam and draft, and assuming an optimal *B/T* around 2.25, it can be demonstrated that the optimum C_M is about 0.80[16], as the wetted surface is getting minimal at this range. However, as the draft is often limited (for large ships and generally for Ro-Ro/RoPax ferry ships, due to the enhanced stability requirements and consequently the *B/T* ratio is larger than optimal), significantly larger C_M than 0.80 results in practice.

[16] It is reminded that the midship sectional coefficient of a half sphere, which is the solid with minimum surface for given volume, is $\pi/4 \cong 0.7854$.

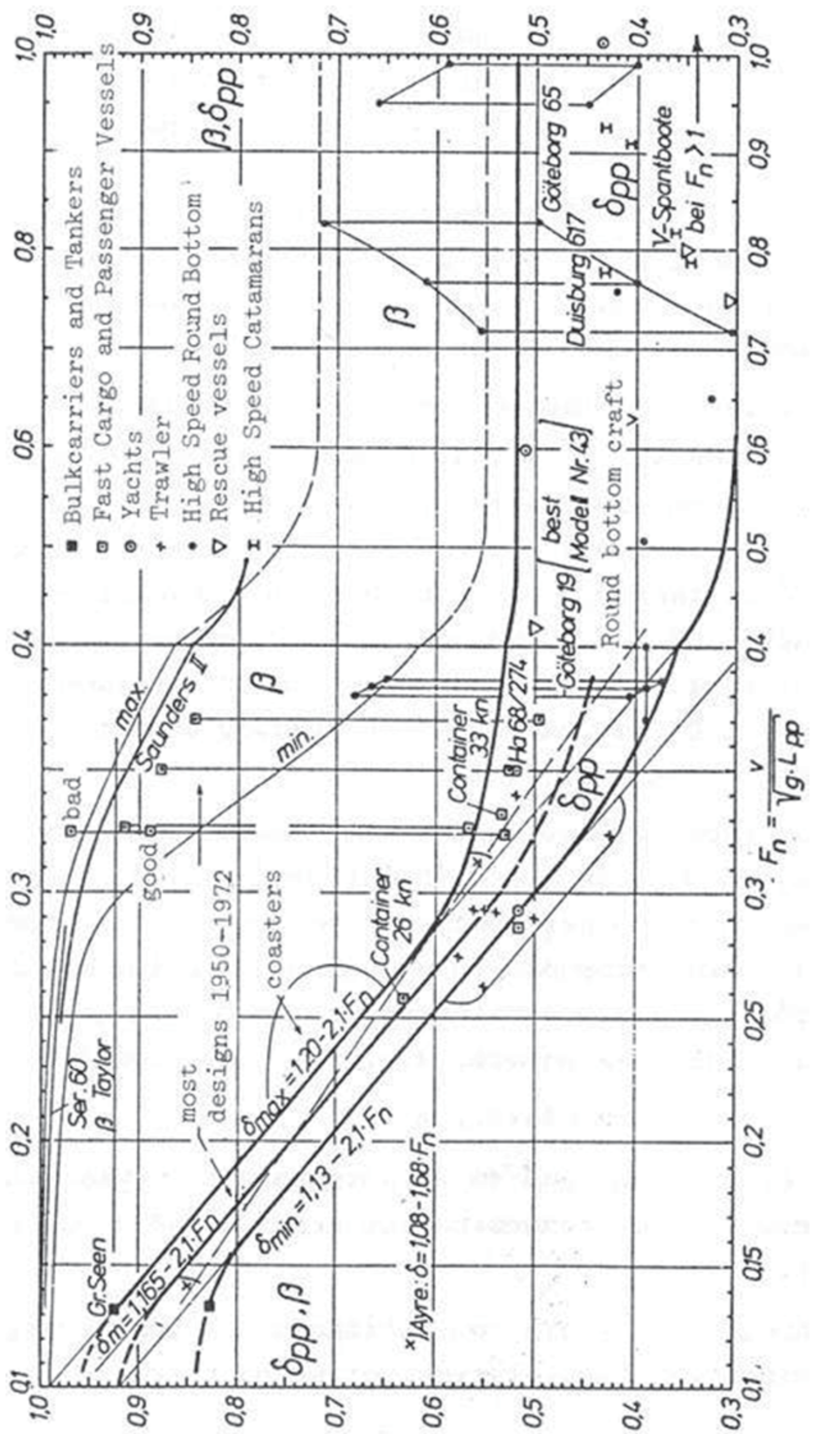


Fig. 2.48 Regions of favorable selection of block coefficient $\delta \equiv C_B$ and midship section $\beta \equiv C_M$

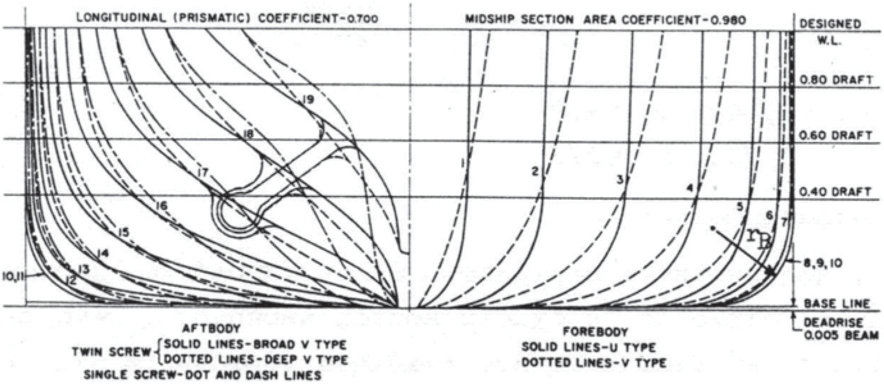


Fig. 2.49 Relationship of bilge radius and deadrise to ship's midship section coefficient

b. Generally for given displacement and main dimensions, i.e., given C_B, an increase of C_M causes an increase of the wetted surface area, lengthening of the transverse flow streamlines, and stronger irregularities in the distribution of the velocity field around the hull, which all contribute to increased frictional and eddy (pressure–viscous) resistance components.
c. On the other hand, for fast ships, where the objective is to minimize the wave-making resistance, it is sought to shift the displacement as downward as possible, even accepting an increase of the local sectional breath over the draft, compared to that at the waterline, thus forming the hull so that eventually $C_M > 1.0$ (for 'bulbous' type sections). Also, due to the increase of the length of entrance of the sectional area curve, for increased C_M and midship sectional area, a wave-making resistance reduction may be expected (see Fig. 2.50).
d. The water flow in the transverse direction especially in the bilge area, is significantly disturbed for large C_M and small bilge radius, resulting in flow separation, generation of eddies, and an increase of corresponding resistance components. Thus for given C_M, it is appropriate to seek a sufficient bilge radius and small deadrise of the bottom.

2.11.2 *Effect on Construction Cost*

The construction effort and particularly the required man-hours for the steel structure manufacturing are reduced in dependence on the extent of fitted flat panels, on the limited number of plates and reinforcements to be bended, and the extent of the ship's parallel body having a constant bilge radius for a certain length of the ship. Thus, a larger possible C_M, small bilge radius (circular instead of parabolic form), and a small or zero deadrise of the bottom, are targeted from the easiness and cost of construction point of view.

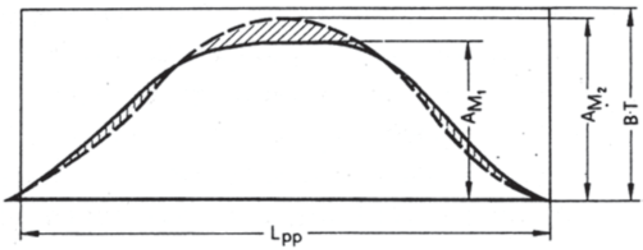


Fig. 2.50 Distribution of sectional area for the same displacement and main dimensions, but different C_M

2.11.3 Effect on Space Exploitation

Especially for ships transporting break or unitized cargo, the demand for larger hold volumes, and flat and rectangular hold surfaces, leads to large C_M coefficients, with a small bilge radius, vertical walls, and long parallel body around amidships.

For containerships, because of the transportation of standard containers in the cells and the unique size of container boxes, it is not recommended to select C_M with criterion the possible fitting in holds of a limited number of additional boxes, which would lead to large C_M values and a small bilge radius; it is rather better to look at the negative effect on the ship's resistance/powering, what is significant for fast[17] cargo ships, like for containerships.

2.11.4 Effect on Stability

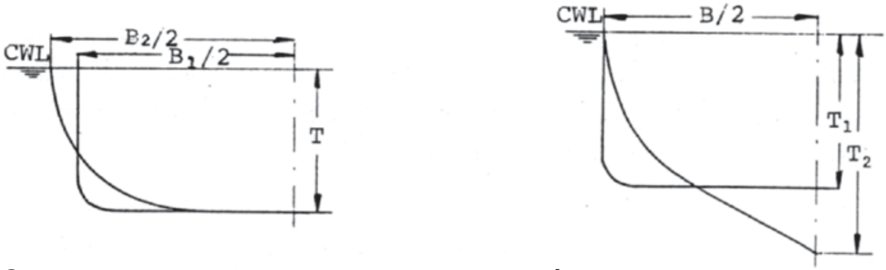
It is possible to increase the initial stability of the ship with an increase of the vertical position of the center of buoyancy and the increase of the breadth on the ship's loaded waterplane. This leads to V-type sections, large drafts, and small C_M coefficients.

If the midship section area A_M is presumed given, then for fixed draft T, an increase of the beam B leads to a reduction of C_M and a significant increase of the initial stability due to the increase of moment of inertia I_T (see Fig. 2.51a).

Also, again for given A_M and fixed beam, increase of the draft T leads to a reduction of C_M and small increase of the initial stability due to the rising of $\overline{KB}$ (see Fig. 2.51b).

Both the aforementioned effects are important for vessels with special stability and propulsion requirements (enabling the fitting of as large as possible propeller

[17] The introduction of "slow steaming" in container shipping in recent years partly affected these considerations; it is noted, however, that despite "slow steaming" in practical operation, containerships continue to be designed as "fast" cargo ships, but taking into account a "slow steaming" operation over certain period of their "life cycle."



a b

Fig. 2.51 Effect of variation of beam and draft on *BM* and *KB*. **a** A_M=const, T=const ($B_1 < B_2$, $C_{M2} < C_{M1}$) ($\overline{BM}_2 > \overline{BM}_1$ $\overline{KB}_2 > \overline{KB}_1$). **b** A_M=const, T=const ($T_1 < T_2$, $C_{M2} < C_{M1}$) ($\overline{BM}_2 = \overline{BM}_1$ $\overline{KB}_2 > \overline{KB}_1$)

diameter that requires large draft) , such as tugboats and fishing ships, for which we observe small C_M coefficients in practice.

2.11.5 Effect on Seakeeping Performance

Generally, ships with small C_M coefficients are sensitive to roll motions due to the reduced damping for the rotational motions about the longitudinal axis. The damping is proportional to the resistance resulting from the transverse water flow and obviously it increases for large coefficients C_M ('squared' sections) and small bilge radius (triggering flow separation).

The normal way of increasing roll damping is to install bilge keels or vertical keels (to small boats), and to larger ships to fit stabilizing fins and antirolling tanks.

The bilge keels' width is usually about or larger than 2 % of the beam of the ship, or 30 % of the bilge radius of the midship section. Their length is about 25 % of the ship's length. The design and proper fitting of the bilge keels are only possible through the conduct of model experiments (or numerical computations CFD) due to the required alignment with the streamlines around the hull so as to avoid the strong increase of pressure–viscous resistance of the vessel when sailing in calm water (Figs. 2.52 and 2.53).

The aforementioned factors, which are in a sense contradictory and mutually exclusive, have in practice led to the following options:

- Generally the C_M coefficient is chosen according to the C_B and decreases for small C_B and high Froude numbers (see Figs. 2.54 and 2.55).
- For small high speed craft with $Fn \geq 0.40$ the C_M may also be reduced for stability reasons.
- Below the limit of $C_M = 0.65$, as shown in Fig. 2.33, it may reach values of 0.50, so as to satisfy the requirements on stability and sufficiency of deck area. This applies, in moderate form, to fishing and tug boats.

Fig. 2.52 Bilge keel on a tug boat

Fig. 2.53 Fin stabilizer on a cruise ship

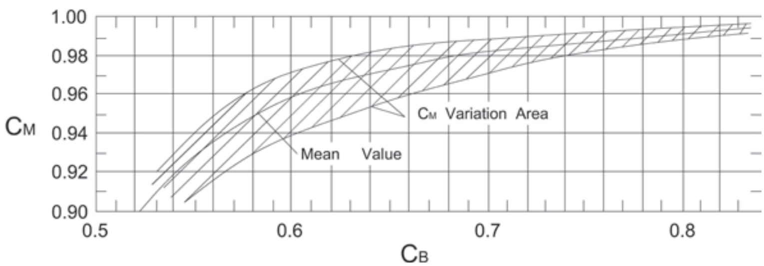


Fig. 2.54 Midship section coefficient versus block coefficient

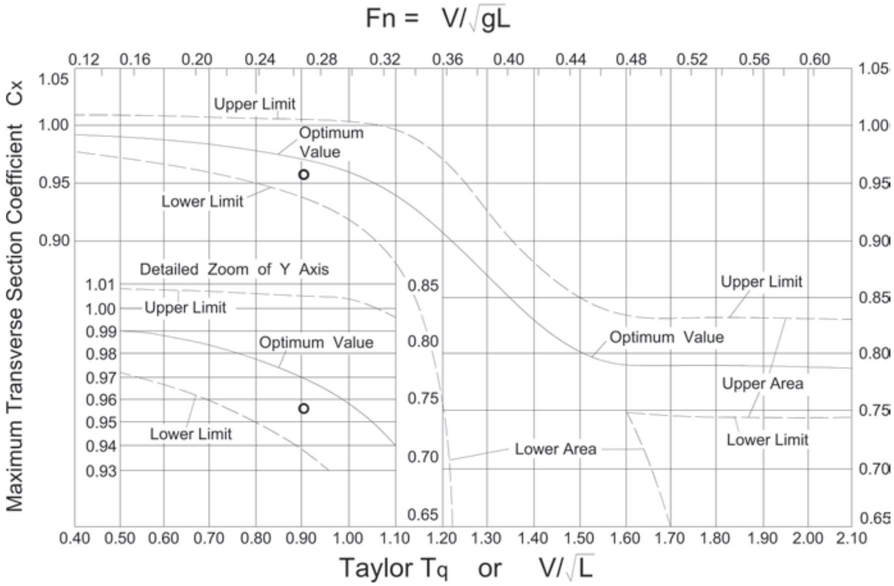


Fig. 2.55 Regions for the selection of the maximum transverse sectional (in general midship) area coefficient versus Froude number or Taylor speed-length ratio

2.11.6 Approximation Formulas

A. Coefficient C_M (large ships without deadrise)
Van Lammeren

$$C_M = 0.9 + 0.1 C_B. \tag{2.90}$$

Kerlen (1979)

$$C_M = 1.006 - 0.0056 C_B^{-3.56} \tag{2.91}$$

Laboratory HSVA (Hamburg)

$$C_M = 1/(1 + (1 - C_B)^{3.5}) \tag{2.92}$$

The above formulas can be applied to relatively large ships with a normal *L/B* ratio.

Tables (see Table 2.6)
Large ships

$$C_M = 0.93 \text{ to } 0.997$$

Small craft (tugs, fishing boats, small ferries)

$$C_M = 0.7 \text{ to } 0.9$$

Table 2.17 Typical sizes of bilge radius and bottom deadrise

	r_B [m]	d_R [m]
Cargo	2.0–2.7	0.0–0.2
Tankers and bulk-carriers	2.0–3.0	0.0
Reefers	2.0–2.7	0.0–0.5
Passenger ships	3.5–5.5	0.0–0.5
Ferries	3.5	0.0–0.6

B. Bilge radius (Schneekluth—without deadrise)

$$r_B = \frac{B \cdot C_K}{\left(\frac{L}{B} + 4\right) C_B^{\,2}}, \qquad C_K = 0.5 \div 0.6.$$

If the above formula is applied to ships with deadrise of height d_R, then the coefficient C_B should be corrected as follows:

$$C_B = C_B \cdot \mathrm{T} / \left(\mathrm{T} - 0.5 d_R\right) \tag{2.93}$$

The following relationship between C_M and r_B (empirical) is valid for ships *without deadrise* (Table 2.17; Fig. 2.56):

$$C_M = 1 - \frac{r_B^{\,2}}{2.33 \cdot B \cdot T} \tag{2.94}$$

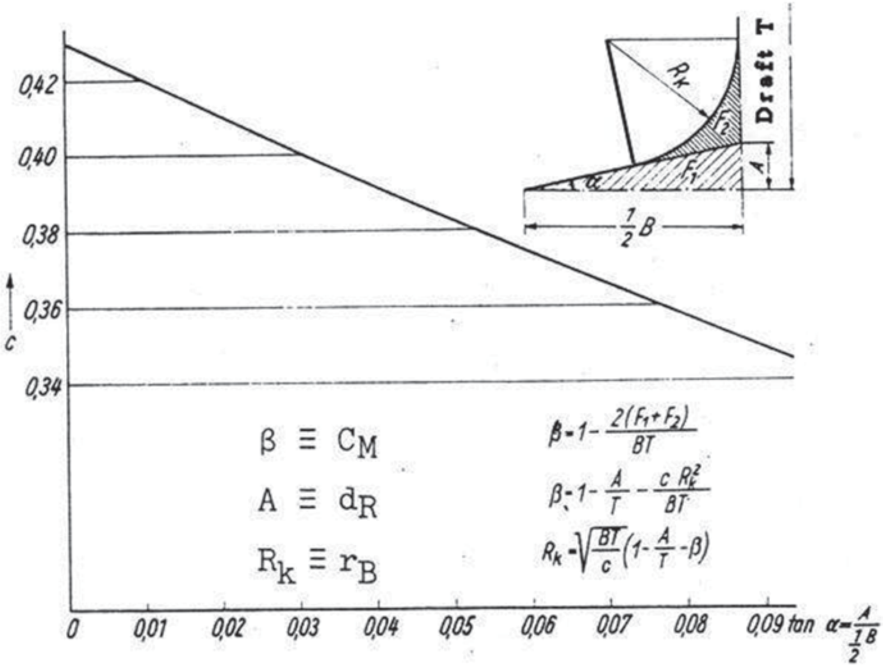


Fig. 2.56 Bilge radius and deadrise according to Henschke (1964)

C. Typical sizes of bilge radius and bottom deadrise

2.12 Waterplane Area Coefficient C_{WP}

The C_{WP} coefficient, which expresses the degree of fullness of the waterplane area in relation to the circumscribed rectangle of length L and width B, is significantly influenced by the form of the transverse sections and by the coefficients C_B and C_M (C_P).

Usually, the C_{WP} coefficient is selected in the preliminary design context so that the stability requirements are satisfied, i.e., namely relatively high C_{WP} values are selected, which affect negatively the ship's resistance (wave-making).

It is however more appropriate to consider in the preliminary selection of C_{WP} values around the lower typical limits and develop the shiplines almost independently from a pre-selected C_{WP} value. This leads to hydrodynamically favorable shiplines, without the C_{WP} value being a constraint for achieving adequate stability. Problems of insufficient stability should be rather treated with more drastic means, for example, change of the main dimensions (beam), of weight distribution, of sectional form character, etc.

2.12.1 *Effect on Stability*

The beam and the waterplane area coefficient influence decisively the calculation of the transverse moment of inertia of the ship's waterplane area, namely, for a given displacement, the magnitude of the vertical distance of the transverse metacenter from the buoyancy center $\overline{BM}$. Accordingly, the length and the C_{WP} coefficient affect the value of the longitudinal metacenter $\overline{BM_L}$.

It is obvious that the moment of inertia of the waterplane area increases as the coefficient C_{WP} increases, likewise the values of $\overline{BM}$ and $\overline{BM_L}$. Meanwhile, assuming constant sectional areas, thus, for given displacement and distribution of it, an increase of C_{WP} leads to V sections with high center of buoyancy, namely to increase of $\overline{KB}$. Overall, an improvement of the *form stability* results, namely of $\overline{KM}$, which is mitigated somewhat by the less pronounced increase of $\overline{KG}$, due to the application V type sections.

The influence of C_{WP} on stability can be approximated as follows: The transverse moment of inertia I_T is considered at first to be known from the formula:

$$I_T = C_{IT} \cdot I_{T*}$$

where

I_{T*}: moment of inertia of the circumscribed rectangle of length L and width B, which is equal to $B^3 \cdot L/12$

C_{IT}: coefficient of specificity of form of waterplane area, $C_{IT} \leq 1.0$.

If we set: $I_T = A_{WP} \cdot r_T^{\,2}$

where A_{WR}: waterplane area, r_T: radius of inertia of waterplane, and consider it according to Hovgaard:

$$r_T = B \cdot (0.0106 + 0.0727 C_{WP})^{1/2},$$

then it is concluded for the coefficient of specificity of form:

$$C_{IT} = C_{WP}(0.1272 + 0.8724 C_{WP}) \tag{2.95}$$

Accordingly it applies to the longitudinal moment of inertia:

$$I_L = C_{IL} \cdot I_{L*} = A_{WP} \cdot r_L^{\,2}$$

where $I_{L*} = B \cdot L^3/12$ and $r_L = L(0.091 C_{WP} - 0.013)^{1/2}$.

Thus, the specificity of form coefficient for the longitudinal moment of inertia is given by:

$$C_{IL} = C_{WP}(1.092 C_{WP} - 0.156) \tag{2.96}$$

The relationship of coefficient C_{WP} with the transverse waterplane specificity of form coefficient C_{IT} is given in the following figures for typical single-screw and multi-screw ships versus the prismatic coefficient C_P (Fig. 2.57).

In the preliminary design stage, the initial stability may be approximated by using the above figures as following.

From the well-known formula of Morrish it shows:

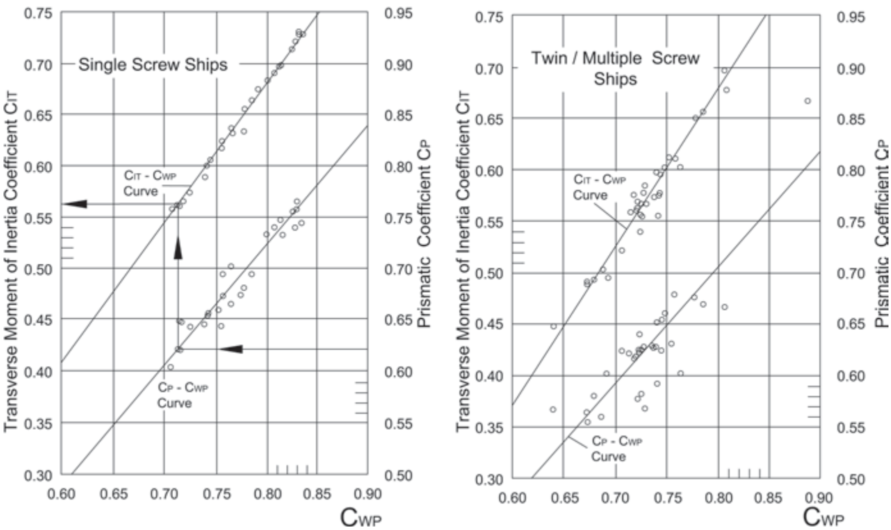


Fig. 2.57 Waterplane area specificity of form coefficient C_{IT} vs. C_{WP} and C_P for single- and twin-screw ships

$$\overline{KB} = T - \frac{1}{3}\left(\frac{T}{2} + \frac{\nabla}{A_{WP}}\right)$$

and based on the approximation of coefficient C_{IT} the moment of inertia I_T is concluded:

$$I_T = C_{IT} \cdot L \cdot B^3 / 12$$

The metacentric radius is determined by:

$$\overline{BM} = I_T / \nabla$$

and consequently the metacentric height:

$$\overline{KM} = \overline{KB} + \overline{BM}$$

Based on the estimation of $\overline{KG}$ (see Sect. 2.15) the resulted $\overline{GM}$ can be evaluated by:

$$\overline{GM} = \overline{KM} - \overline{KG}$$

which should not be smaller than about 0.06B in general, whereas other more specific stability criteria also apply regarding the min *GM* value (see Sect. 2.18).

2.12.2 Effect on Resistance, Propulsion, and Seakeeping Performance

The influences of C_{WP} on the various aspects of the ship's hydrodynamic performance (resistance, propulsion, behavior in waves) are diverse and complicated.

For relatively slow ships, high prismatic coefficient values and an almost evenly, lengthwise distributed displacement lead to a center of buoyancy (and center of flotation in general) forward of amidships; the waterplane lines are very full, especially forward of the midship section. The local waterplane area coefficient, forward of midship, can reach values of 0.90 to 0.95 (tankers and bulkcarriers). In this way the wetted surface of the ship's hull is minimized, for given displacement, and the frictional resistance is reduced.

On the other hand, abaft the midship section, the fullness of the waterplane lines and the local C_{WP} value declines (to about 0.80) so as to achieve a favorable flow to the propeller and to avoid strong flow separation (which increases the eddy/pressure viscous resistance) .

For relatively fast ships, with a significant percentage of wave-making resistance, high C_{WP} values will lead to the generation of intense local waves at both the entrance and the run of the waterlines, as well as around the shoulders. An

extremely sharp waterline at the ends is favored to avoid intense waves in the bow and stern region; however, this may result particularly to more pronounced local waves around amidships. Generally, for a given speed (Froude number) and beam, the optimal C_{WP} values decrease with the increase of Froude number.

As to the influence of C_{WP} on the ship's performance in waves (motion amplitudes and phases, added resistance in waves), it has been observed in experiments and computations that high C_{WP} values, i.e., very full waterplane lines at the bow, have negative influence on seakeeping, especially on the sailing of the ship in head seas due to likely slamming problems etc.

2.12.3 Approximation Formulas

In general, the waterplane fullness coefficient C_{WP} is a function of block coefficient C_B and of the character of the ship's sections. Special types of ships with a large *L/B* ratio are likely to have both U and V sections, whereas a small *L/B* ratio is mainly associated with intense V sections. In addition, ships with relatively small *B/T* ratio are combined with high C_{WP} values to achieve sufficient stability and deck area.

The basic empirical formulas for the approximation of C_{WP} in the preliminary design phase are:

Intense U type sections

$$C_{WP} = 0.95C_P + 0.17(1 - C_P)^{1/3}\,\text{(Schneekluth)} \tag{2.97}$$

or

$$C_{WP} = 0.778C_B + 0.248$$

Normal sections

$$C_{WP} = (1 + 2C_B)/3 \tag{2.98}$$

Intense V type sections

$$C_{WP} = (1 + 2C_B/C_M^{0.5})/3\,\text{(Schneekluth)} \tag{2.99}$$

or

$$C_{WP} = 0.793C_B + 0.297.$$

The formulas are applicable at first only to ships with cruiser stern. Ships with transom stern generally have higher C_{WP} values. For ships with significant overhang of the wetted part of the stern beyond the aft perpendicular, the correction of the C_{WP} with the following coefficient is applied:

$$K = 1 + C_{\mathrm{P}} \cdot (0{,}975 L_{\mathrm{WL}} - L_{\mathrm{PP}})/L_{\mathrm{PP}} \tag{2.100}$$

2.12.4 Conclusions

a. To achieve satisfactory *form* stability, the increase of beam B should be preferred, which affects more drastically the moment of inertia of the ship's waterplane area, rather than the C_{WP}.
b. In the preliminary design phase and when using approximate formulas the selection of high C_{WP} values must be avoided, as these values may be reduced in the course of the ship's design (development of ship lines), resulting in poor stability.
c. In the transom stern case, which is always accompanied by high C_{WP} values, it needs to be considered that a possible stern emergence due to trim, motions in waves, etc., <u>will</u> cause a considerable loss of waterplane area and hence of stability (drastic $\overline{GM}$ reduction). In specific seaway conditions (following and head seas), this may lead some ships to severe roll motions (*Mathieau instabilities and parametric roll phenomena*) .

2.13 Determination of the Main Dimensions Through the Ship Design Equation

The "design equation" (in German *Entwurfsgleichung,* Schneekluth 1985) leads to the determination of the main dimensions of a study ship through the selected ratios of main dimensions and form coefficients of similar ships. In case of lack of data from similar ships, then empirical formulas and data from empirical diagrams, which are supposed to be applicable to the current ship type, can certainly be used.

The "design equation" is derived from the already known "displacement equation" (see Appendix C). As is well known, it holds for the displacement (weight):

$$\Delta = \rho_{\mathrm{SW}} \cdot g \cdot \nabla^* \tag{2.101}$$

where

ρ_{SW}: density of sea water
Δ^*: volume of displaced water $= C_{\mathrm{B}} \cdot L \cdot B \cdot T \cdot k_{\mathrm{A}}$
k_{A}: coefficient of correction of the displaced volume (design—molded volume) for average shell thickness, appendages, etc. (see Sect. 2.15).

Introducing the ratios *L/B* and *B/T*, which, as known, significantly influence both the ship's resistance (*L/B*) and stability (*B/T*), the form of the displacement equation can be rearranged as follows:

$$\Delta = \rho_{\mathrm{SW}} \cdot g \cdot (L/B) \cdot B^2 \cdot [B/(B/T)] \cdot C_{\mathrm{B}} \cdot k_{\mathrm{A}}$$

or

$$\Delta = \rho_{\mathrm{SW}} \cdot g \cdot C_{\mathrm{B}} \cdot [(L/B)/(B/T)] \cdot B^3 \cdot k_{\mathrm{A}}$$

Thus, the following expression is concluded for the beam:

$$B = \left[\frac{\Delta \cdot (B/T)}{\rho_{\text{SW}} \cdot g \cdot C_B \cdot (L/B) \cdot k_A}\right]^{1/3} \tag{2.102}$$

thus, the beam is the only unknown in the above displacement equation, assuming the right hand side known.

Likewise, for known (*L/B*) and (*L/T*) ratios from similar ships, the following expression for the ship's length is concluded:

$$L = \left[\frac{\Delta \cdot (L/B) \cdot (L/T)}{\rho_{\text{SW}} \cdot g \cdot C_B \cdot k_A}\right]^{1/3}$$

As mentioned earlier (see Sect. 2.1), for *deadweight carriers* the estimation of displacement Δ through the transport capacity (DWT) is readily possible; also, the ratios (*B/T*), (*L/B*) and (*L/T*) and C_B coefficient can be estimated from data of similar ships.

For *volume carriers* the above methodology may be modified by including (instead of the displacement) the *underdeck-volume*, ∇_D namely the ship's displaced volume up to a waterline at the height of the main deck:

$$\nabla_D = C_{BD} \cdot L \cdot B \cdot D$$

where C_{BD}: hull coefficient for a waterline at the height of *D* (main deck) (see Sect. 2.15.4 approximation formulas, function of C_B).

Thus, we have for the beam

$$B = \left[\frac{\nabla_D \cdot (B/D)}{C_{BD}(L/B)}\right]^{1/3} \tag{2.103}$$

Assuming that the required volume ∇_D can be estimated for the volume carriers (see Sect. 2.17.2), the further process resembles the previously described for deadweight carriers, provided that the ratios (*B/D*), (*L/B*) and the C_{BD} are known from similar ships.

2.14 Preliminary Estimation of Propulsive Power

During the ship's conceptual/preliminary design, the *exact* knowledge of the required propulsive power for achieving the speed specified in owner's requirements *is not required*; this also applies to the other hydrodynamic ship characteristics, which relate to the selection of the propeller and the rudder.

In a ship's initial design phase, which eventually aims at a first approximation of the ship's total weight (including the weight of the machinery installation and the approximate required engine room volume) and of the corresponding displacement of the ship, *a preliminary estimation* of the ship's propulsive power is enough for the calculation of the weight (and engine room volume) of the propulsion plant and

fuel. This approximation can be based on empirical formulas, data of similar ships or diagrams deduced from statistical data for various types and sizes of ships.

Commonly used approximate methods[18] for the estimation of the preliminary propulsive power P (installed power) of the ship are:

a. **British Admiralty formula**

$$C_{\mathrm{N}} = \frac{\Delta^{2/3} V^3}{P} \tag{2.104}$$

where

Δ: displacement [t], V: speed[kn], P: installed power in [HP] or [kW].

The Admiralty constant C_{N} can be calculated from data of similar (parent) ships based on the *same reference units* for Δ, V and P, [tons], [kn], and [HP] or [kW]. In the use of this method it is tacitly assumed that the *parent* (*similar*) ships have similar hull form and not significant differences in the Reynolds and Froude numbers (i.e., *the length and speed must be about the same*). The formula can be used for the estimation of the brake horsepower P_{B}, or shaft power P_{S} or delivered or effective power, depending on the availability of data from the parent ship; also, the constant can be given in other units, for example, V[m/s], P[kW], assuming that C_{N} is appropriately defined and used.

Variation of the Admiralty formula by Völker (1974):

$$P_{\mathrm{D}} = \frac{\Delta^{0.567} V^{3.6}}{1671 \cdot \eta_D} \cong \Delta^{0.567} V^{3.6} \cdot 10^{-3} \tag{2.105}$$

where Δ[t], V[kn], P[kW] (see also Fig. 2.58 by Völker (1974), P_{D}[HP]).

A similar to the British Admiralty constant was more recently introduced by Heickel (Papanikolaou 2002):

$$K = (\sqrt{\nabla} / P_{\mathrm{B}})^{1/3} \cdot V_{\mathrm{T}}, \tag{2.106}$$

where Δ is the displaced volume in m³, V_{T} the trial speed in [m/s] and P_{B} the brake horsepower in [kW] (Fig. 2.59).

[18] The following semi-empirical methods proved in practice satisfactory for the for more precise calculation of the total resistance and powering of common types of ships in the preliminary design phase:

Holtrop, J., Mennen, G. G. J., "An Approximate Power Prediction Method," Journal International Shipbuilding Progress, 29(335), July 1982.

Holtrop, J., "A Statistical Re-analysis of Resistance and Propulsion Data", Journal International Shipbuilding Progress, 31(363), November 1984.

Hollenbach, U., "Estimating Resistance and Propulsion for Single-Screw and Twin-Screw Ships in Preliminary Design", Proc. of the 10th ICCAS Conference, Cambridge, MA, June 1999.

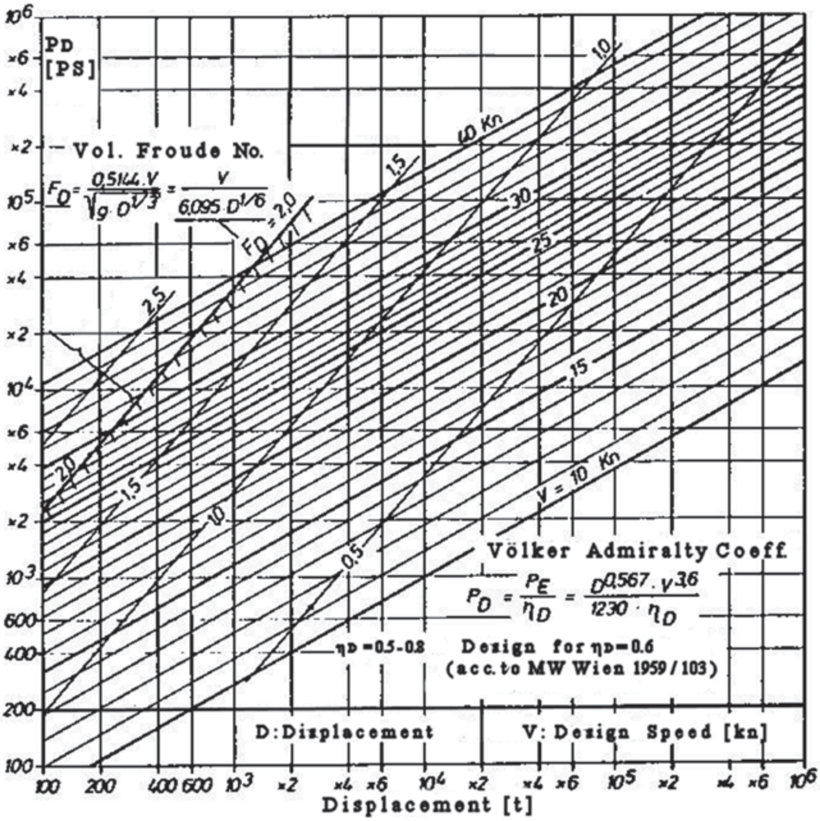


Fig. 2.58 Approximation of propulsion power versus displacement and volumetric Froude number by Völker (1974)

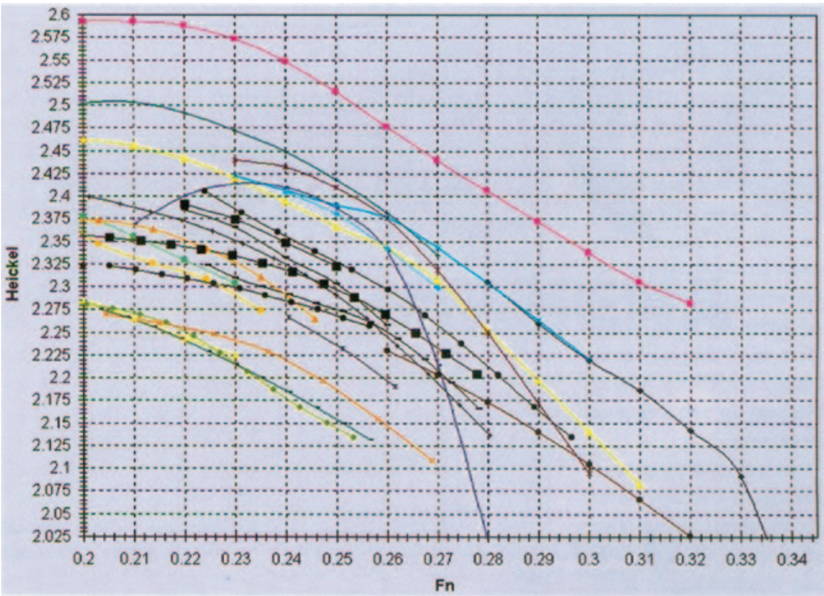


Fig. 2.59 Heickel Coefficients for modern Ro-Ro ships according to Deltamarin. (Papanikolaou 2004)

b. **Use of diagrams and empirical formulas**

The use of the following empirical diagrams by MAN B&W and Harvald is recommended for dry cargo and liquid cargo ships with common type of propulsion plants (Figs. 2.60, 2.61, and 2.62).

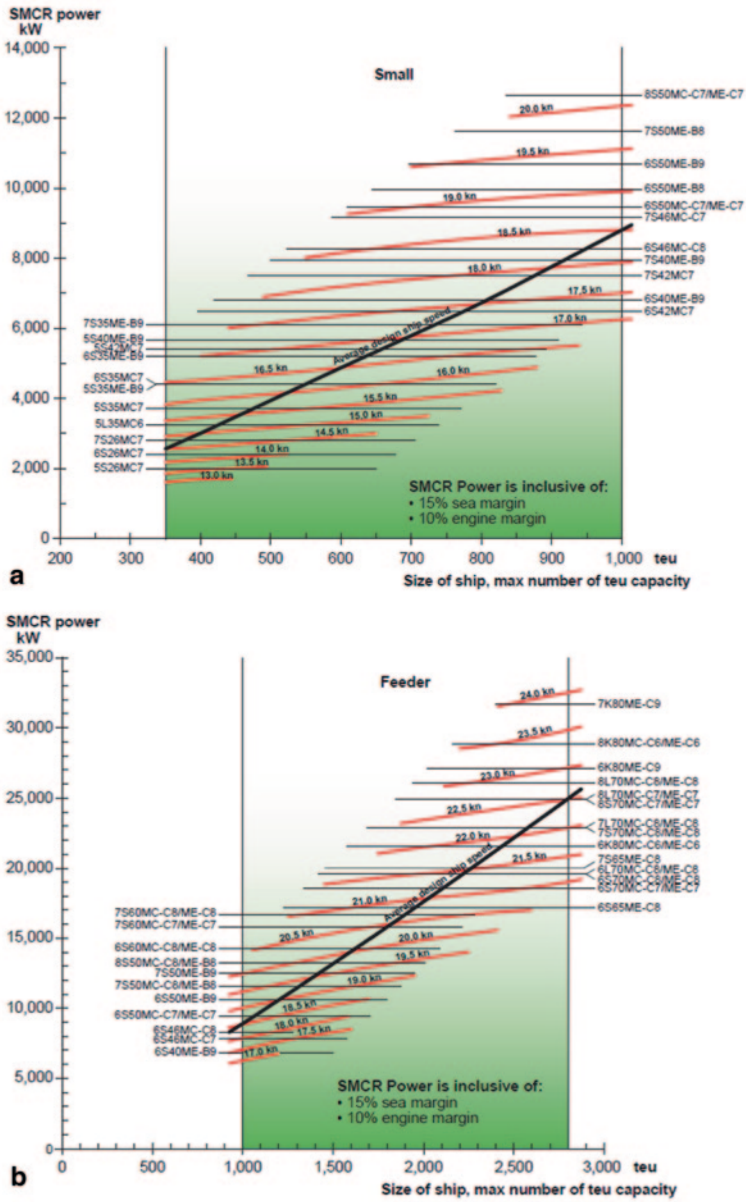


Fig. 2.60 Diagrams of installed propulsion power for containerships versus DWT and speed V [knots]

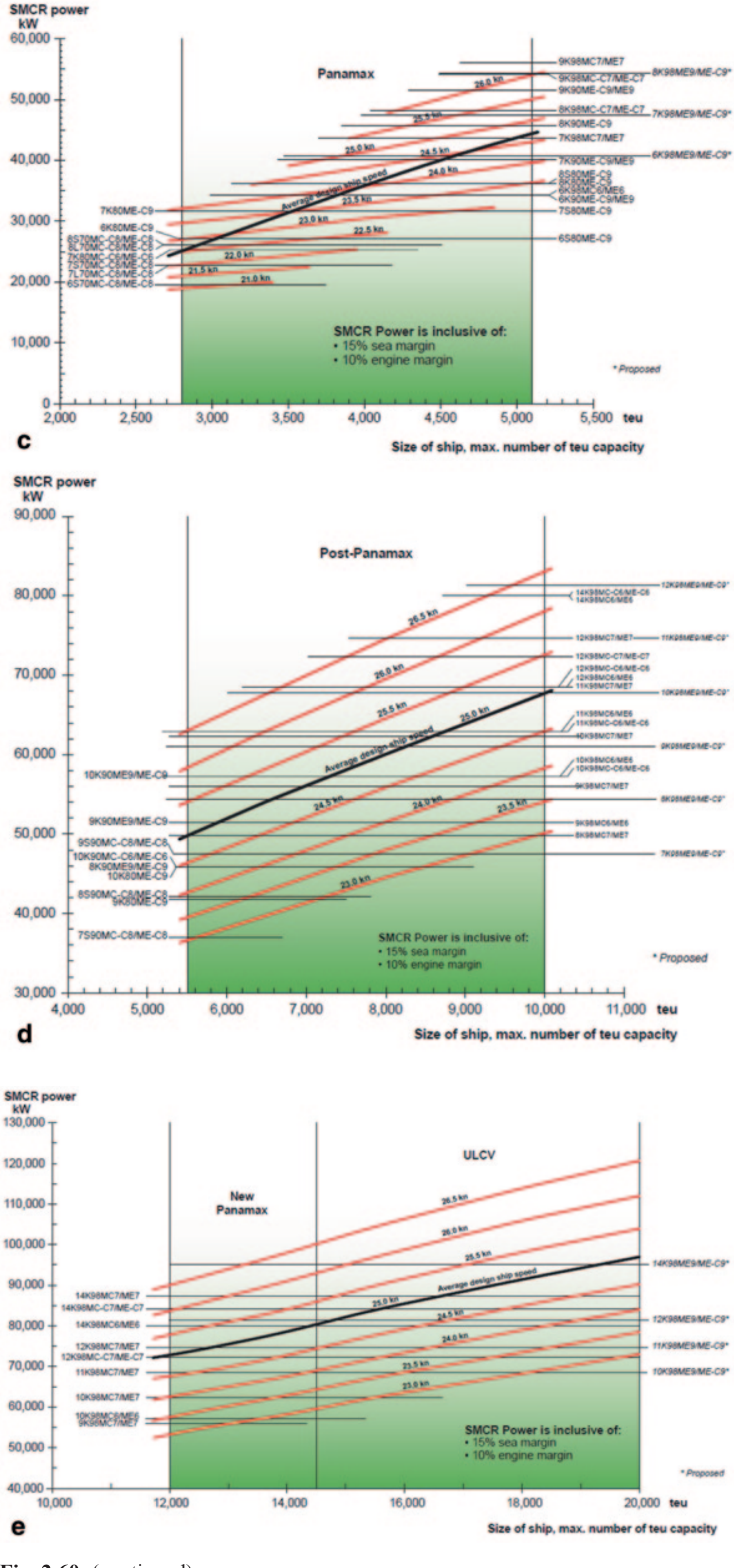


Fig. 2.60 (continued)

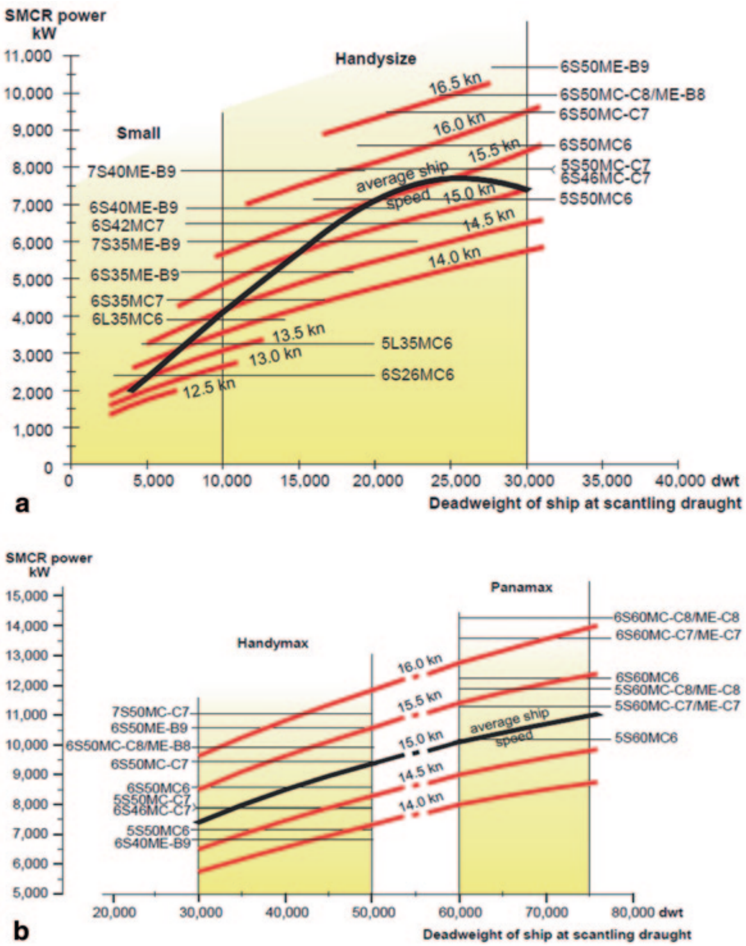


Fig. 2.61 Diagrams of installed propulsion power for tankers versus DWT and speed *V* [knots]

b1. **Estimation of the installed horse power of modern ships by MAN B&W Diesel A/S (2005)**

Figs. 2.60	a, b, c, d, e	$SMCR = f(TEU, V)$,	Container ships
Figs. 2.61	a, b, c, d	$SMCR = f(DWT, V)$,	Tankers
Figs. 2.62	a, b, c	$SMCR = f(DWT, V)$,	Bulk carriers

where SMCR: specified maximum continuous rating

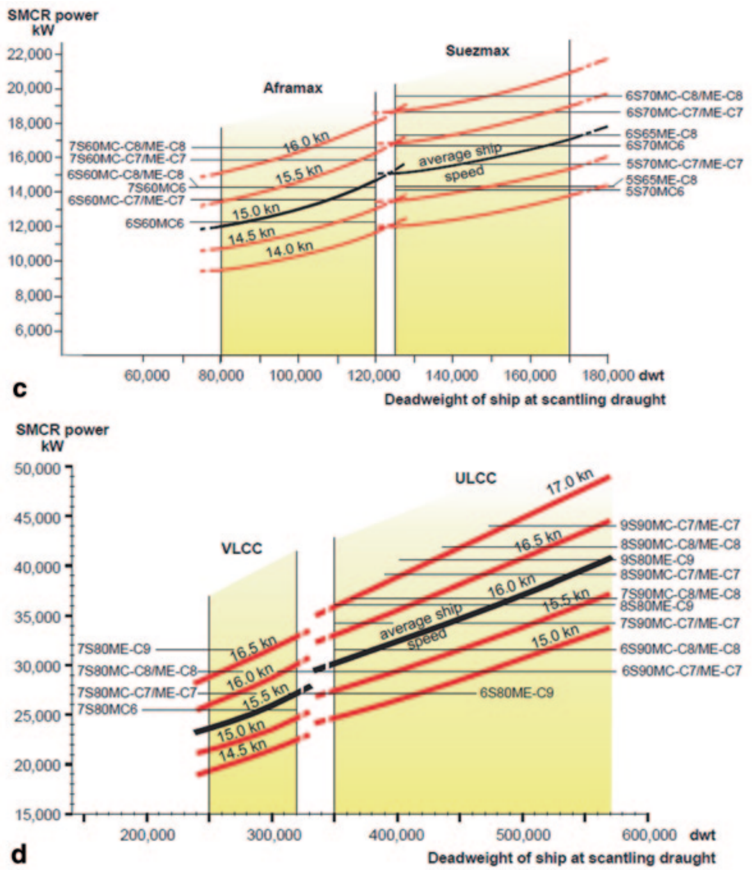


Fig. 2.61 (continued)

b2. **Estimation of the installed horse power of ships according to Harvald**

$$\text{Fig. 2.63} \quad P_B = f\left(\Delta, V, L/\nabla^{1/3}\right), \quad C_B = 0.60$$

$$\text{Fig. 2.64} \quad P_B = f\left(\Delta, V, L/\nabla^{1/3}\right), \quad C_B = 0.70$$

$$\text{Fig. 2.65} \quad P_B = f\left(\Delta, V, L/\nabla^{1/3}\right), \quad C_B = 0.80$$

where P_B: break horse power

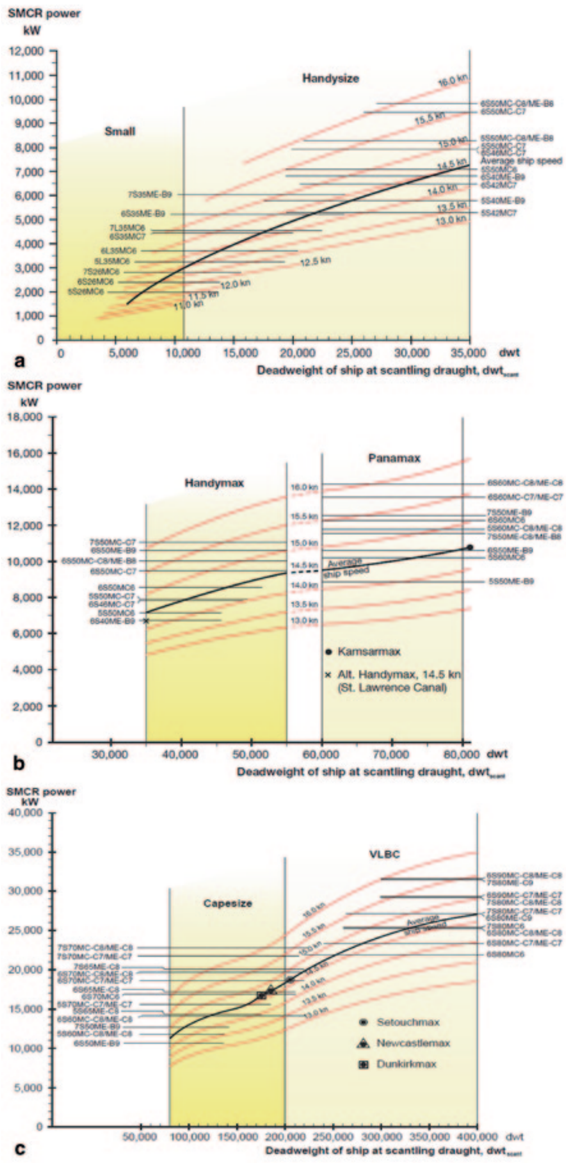


Fig. 2.62 Diagrams of installed propulsion power for Bulk Carrier versus DWT and speed V [knots]

Limits of parameters

TEU	=(400) to 18,000
DWT	=(2,000) to 580,000 t
V	=(11) to 26.5 knots
Δ	=(100) to $100 \cdot 10^4$ t (Figs. 2.63, 2.64, and 2.65)
$L/\nabla^{1/3}$	=4.0, 5.0, 6.0 (Figs. 2.63, 2.64, and 2.65)
C_B ($\equiv \delta$)	=0.6, 0.7, 0.8 (Figs. 2.63, 2.64, and 2.65)

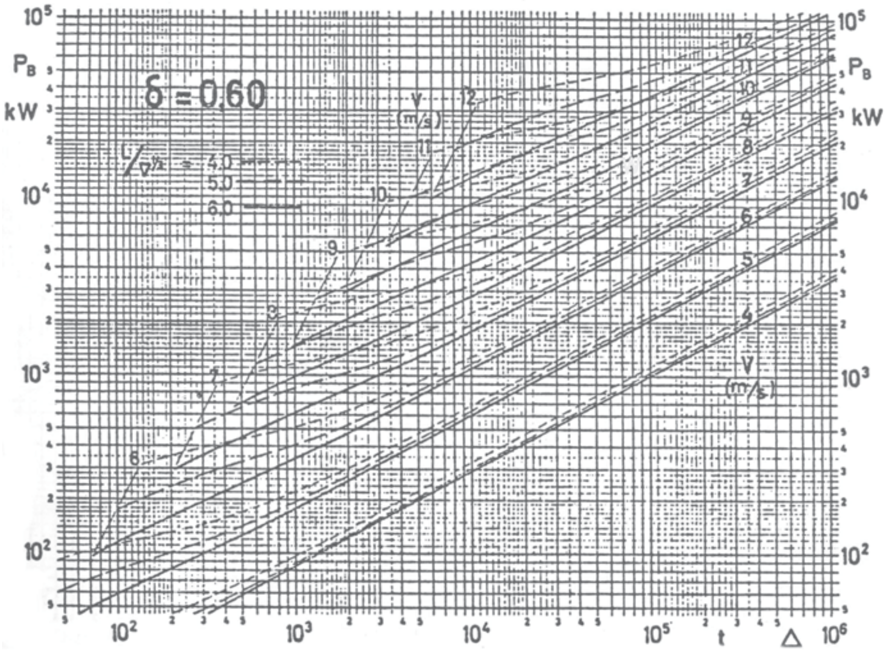


Fig. 2.63 Diagrams of installed propulsion power [kW] versus the displacement Δ [tons], velocity V [m/s] and slenderness ratio $L/\nabla^{1/3}$, $C_B = 0.60$ acc. to Harvald

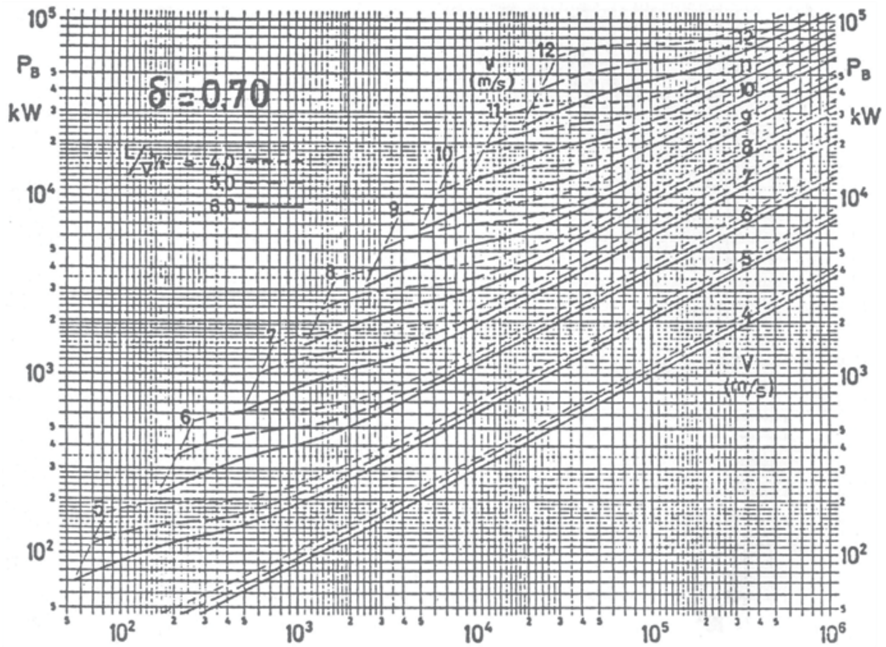


Fig. 2.64 Diagrams of installed propulsion power [kW] versus the displacement Δ [tons], velocity V [m/s] and slenderness ratio $L/\nabla^{1/3}$, $C_B = 0.70$ acc. to Harvald

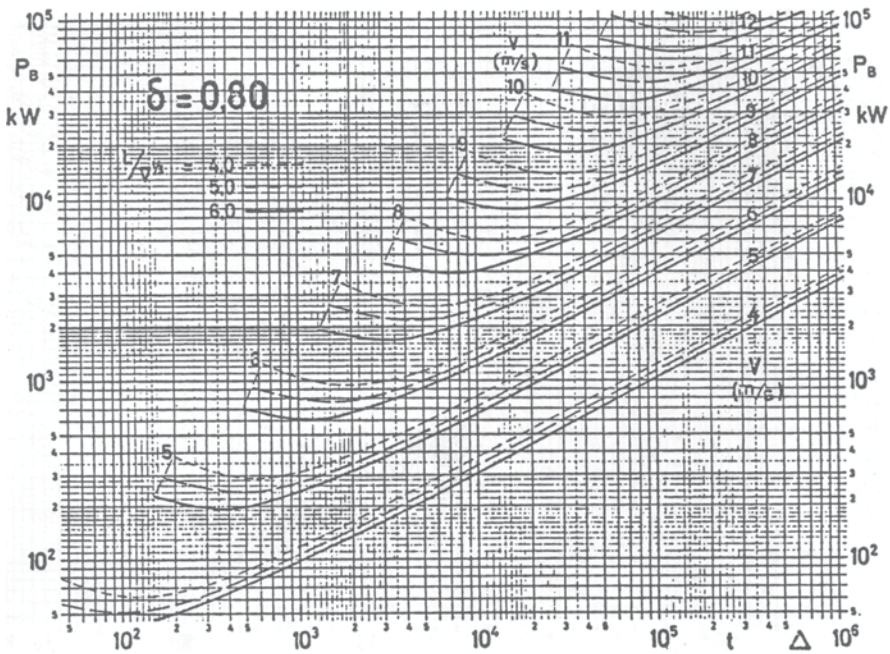


Fig. 2.65 Diagrams of installed propulsion power [kW] versus the displacement Δ [tons], velocity V [m/s] and slenderness ratio $L/\nabla^{1/3}$, $C_B = 0.80$ acc. to Harvald

2.15 Estimation of Ship Weights

The as accurate as possible approximation of the various weight groups of the ship, and the position of their centroid, is a very important step in both the preliminary and the final ship design stage. Likewise, any inaccuracy and mistakes have significant influence on the achieved transport capacity, as on the speed, stability, and safety of the ship[19]. Also, due to the indirect association of the ship's construction cost with the acc. to Harvald ship weight, particularly the structural steel weight, the as possible accurate assessment of the various weight groups is already of great importance in the preliminary design phase, because it concerns the terms of the initial tender of a shipyard to the interested shipowner.

2.15.1 Definitions of Ship Weight Components

The displacement equation may be analyzed as following:

$$\Delta = W = W_L + \mathrm{DWT} \tag{2.107}$$

Δ: displacement (weight of displaced water)
W: total, sum of weights of the ship (weight)
W_L: weight of light(empty) ship (sometimes LS)
DWT: transport capacity, deadweight.

a. **Analysis of light ship weight W_L**

Definition of W_L It corresponds to the weight of the finished, fully equipped, and seaworthy ship *without* supplies and payload. In this weight the following machinery supplies are included: lubricants and cooling water of machines, feed water of boilers, weight of liquids in pipes. The weight W_L corresponds roughly to the ship's delivery state from the shipyard to the shipowner.

Analysis of W_L

$$W_L = W_H + W_M + R \tag{2.108}$$

where

W_H weight of hull,
W_M weight of machinery
R reserve (margin/ tolerance of estimations)

[19] Whereas small inaccuracies in the estimation of ship's weight may be balanced by slight changes of ship's draft, this is very different when dealing with the proper estimation of weights of submarines, as there the imbalance of the sum of weights and displaced volume trivially leads to submarine's inability to float in neutral equilibrium. Additionally, it must be ensured that in all cases the center of the overall mass must be below the center of displaced volume for the submarine to be stable (have positive stability).

Analysis of W_H The hull weight W_H can be further broken down into:

$$W_H = W_{ST} + W_{OT} \tag{2.109}$$

where:

W_{ST}: weight of steel structure
W_{OT}: weight of outfitting

Definition of W_{ST} It includes the weight of all elements of the steel structure of the ship and corresponds approximately to a shipyard's steel work. In addition to all the plates and stiffeners of the ship, the following components are included in this weight group as well: the mounting base of the engine, the superstructure and deckhouses, even if they are of different materials (e.g., aluminum), the masts, the rudder, the rudder shaft, the hatch coamings, the bulwark.

Definition of W_{OT} It includes the weight of all fittings to the "naked" ship and also all detachable outfittings of the ship *except for the machinery outfitting (*see Table 2.30) for description of elements of W_{OT}). Certain elements of the W_{ST} can be taken as well within W_{OT}, for example, the masts and the rudder, noting that it depends on the practice of the shipyard or designer.

Analysis of W_M:

$$W_M = W_{MM} + W_{MS} + W_{MR} \tag{2.110}$$

where

W_{MM}: main machinery weight
W_{MS}: shaft of propeller and propeller weight
W_{MR}: rest machinery weight

Definition of W_{MM} It includes the weight of the main engine and gearbox (if any), for turbine driven ships the weight of the turbine, the gearbox and boilers respectively.

Definition of W_{MR} It includes the weight of pumps of any kind, any piping inside the engine room, funnels, main electric generators (the emergency electric generator is very often included in W_{OT}), transformers and switchboards, any support mechanical components of the main engine, etc.

Definition of R The reserve (tolerance/margin of uncertainty) R is set in the preliminary design to cover possible inaccurate initial approximations of the various weight groups. Typical values of R, in the preliminary design stage in [%] W_L, are 1–2 % for simple structures (tankers and bulkcarriers) and 2–3 % (up 6 % according to Schneekluth) for more complex ships. With the progress of the design the reserve R diminishes and converges to the *tolerance of construction*, which covers the

differences with respect to the estimated weight of the processed materials and outfitting coming from external suppliers or which are produced by the shipyard itself. During the final phase of the design, the value of R is 0.5–1 % for simple ships and 1–2 % of W_L for complex ones (e.g., passenger ships, reefers, containerships, etc.).

As to the impact of the center of gravity/mass of R on stability, it may be assumed that the vertical position of the weight center of R is *located 20 % higher than the estimated* $\overline{KG}$ *of the vessel*, but the longitudinal position is assumed the same as the estimated longitudinal gravity/mass center of the ship.

b. **Analysis of deadweight DWT**

$$\mathrm{DWT} = W_{\mathrm{LO}} + W_{\mathrm{F}} + W_{\mathrm{PR}} + W_{\mathrm{P}} + W_{\mathrm{CR}} + B \tag{2.111}$$

where,

W_{LO}: weight of the payload (for cargo ships: cargo payload, for Ro-Ro ships: weight of carried vehicles)
W_{F}: fuel weight, including fuel reserve and lubricants
W_{PR}: weight of provisions and water supplies
W_{P}: weight of passengers and luggage (persons & effects); cargo ships may carry up to 12 passengers; for passenger ships, this weight may be included in the payload
W_{CR}: weight of crew (including their luggage)
B: weight of *nonpermanent* ballast (water), whenever is required in the *full load* condition (design draft)

2.15.2 Initial Estimation of Weights and Their Centroids

During the initial estimation of displacement, especially when it comes to cargo ships (dry or liquid cargo), it is possible to approximate the weight of lightship W_L, or the ratio (DWT/Δ), through coefficients, which are dependent on the ship type, Froude number, and the size of the ship (in terms of transport capacity). Such relationships are well known for long time (e.g., Völker (1974), or see Table 2.1), but they are not recommended for volume carrier ships, where the decisive elements of the ship size are the large deck areas, extended large superstructures, or high horsepower, as happens with passenger ships, ferries, tug boats, having all a small (DWT/Δ) ratio.

Typical values of (DWT/*D*) are given in Table 2.1 (Sect. 2.1) for various types ships according to Schneekluth (1985) and others, as well as given in other course supporting material of the author (Papanikolaou and Anastassopoulos 2002).

Regarding the initial estimation of the vertical position of the mass center of the fully loaded ship, the use of the following relationship between $\overline{KG}$ and the side depth D is proposed:

$$\overline{KG} = C \cdot D_S \tag{2.112}$$

where the modified side depth D_S is defined as

$$D_S = D + \nabla_{SS}/(L_{PP} \cdot B) \tag{2.113}$$

and ∇_{SS}: volume of superstructures and deckhouses.

The C coefficients may be taken according to Dudszus and Danckwardt (1982) as the following typical values (Table 2.18):

As to the vertical position of center of gravity of the various groups of weights, the following data of Table 2.19 according to Schneekluth (1985) can be used.

Likewise, in the *support material to* the course *Ship Design and Outfitting I* (Papanikolaou and Anastassopoulos 2002) approximate values for the vertical and longitudinal position of the centers of various groups of weights and types of ships are given according to E. Strohbusch (1971).

2.15.3 Factors That Affect the Values of the Weight Coefficients

When using empirical coefficients for the approximation of the various weight categories, see Sect. 2.15.2, we must pay attention to the indicated upper and lower limits of the magnitudes in Tables 2.18 and 2.19, as well as to the specific features of the concerned ship in the context of the same ship category. For the proper selection of coefficients, it is not sufficient to use average values between the given limits; instead, the following criteria must be taken into consideration:

a. General effects regardless of ship type

a1. *Absolute size*: With the increase of the absolute size of a type of ship, generally the weight coefficients of the ship *decrease* due to the following reasons:

- All structural elements that support local loads remain the same and therefore smaller ships are charged proportionally with more steel weight,
- Generally areas/surfaces increase with $\Delta^{2/3}$
- The number of crew and the extent of their accommodation increase slightly or not at all, when increasing the ship's size (stepwise change)
- The propulsive power increases with $\Delta^{2/3}$

Table 2.18 Coefficients C for the estimation $\overline{KG}$ for various ship types

Passenger ships	0.67–0.72
Large cargo ships	0.58–0.64
Small cargo ships	0.60–0.80
Bulk carriers	0.55–0.58
Tankers	0.52–0.54
Fishing vessels	0.66–0.75
Tug boats	0.65–0.75

Table 2.19 Vertical position of center of gravity of weight groups W_{ST}, W_{OT}, W_M, W_L for the main types of commercial ships as a percentage [%] of the corrected side depth (strength deck) D_S—Synthesis of data by H. Schneekluth (1985)

Ship type	Lower limit[a]	W_{ST}	W_{OT}	W_M	W_L
Cargo ships	5,000 t DWT	60–68	110–120	45–60	70–80
Coastal cargo ships	499 GRT	65–75	120–140	60–70	75–87
Bulkcarriers	20,000 t DWT	50–55	94–105	50–60	55–68
Tankers	25,000 t DWT	60–65	80–120	45–55	60–65
Containerships	10,000 t DWT	55–63	86–105	29–53	60–70
Ro-Ro	$L \cong 80$ m	57–62	80–107	33–38	60–65
Reefers	300,000 ft^3	58–65	85–92	45–55	62–74
RoPax ferries		65–75	80–100	45–50	68–72
Trawlers	$L \approx 44$ m	60–65	80–100	45–55	65–75
Tug boats[b]	$P_B \cong 500$ kW	100–140	70–80	60–70	70–90

[a] Smaller ships within the same category (lower limit) generally have higher positions of centers of weights
[b] For the tugboats the upper values correspond to vessels with extended forecastle

Thus, for example, a large tanker will be generally having values in the lower limits of the cited weight coefficients, while the opposite holds for a smaller one. Of course, this is not the general rule for all types of ships. For example, larger multipurpose cargo ships may dispose *additional* cargo handling facilities and equipment (derricks/cranes of heavy lift capacity, reefer spaces, etc.), thus they may be proportionally heavier than smaller ones.

a2. *Effects on steel weight*:

- Through the exploitation of developments of technology and of computational/optimization methods regarding the ship's structural design, modern shipbuildings are generally lighter than the corresponding older ones with comparable capacity/specifications. It should be noted, however, that for some types of ships (such as tankers), the development of more stringent safety regulations over the years (in particular the marine environment protection regulations, MARPOL and OPA90 introducing *double-hull concept for tanker ships*) led to increased steel weight requirements, for tankers of the same transport capacity. It may be anticipated, however, that increased requirements and savings through optimization and new technologies acted counterbalancing in the historical development of the structural steel weight of tankers.
- The use of lightweight materials is notable, especially in the superstructure of passenger ships; also, the increased use of higher tensile steel (particularly in high stress areas of the structure of large tankers, bulkcarriers and containerships) led to a relative reduction of structural weights for many ship types.

Table 2.20 Effect of speed on the light ship weight. (Strohbusch 1971)

	Cargo ship		Tanker		Bulk-carrier	
F_n	0.18	0.25	0.16	0.19	0.18	0.21
W_L/Δ [%]	28	38	16	24	24	27
W_L/LBD [kp/m³]	150	190	110	140	130	155

The given data refer to relatively old shipbuildings from the 70s and are of interest only in view of the *qualitative* effect of changing the concerned parameters. Generally, the weight coefficients of the light ship have reduced significantly over the years due to optimization of the steel weight and the use of higher tensile steel, especially for tankers and bulkcarriers. Indicative values for modern tankers of double-hull concept are, see Lamb (2003): W_L/Δ [%], $W_L/L\,B\,D$ [kp/m³], F_n [-]=23.3, 119, 0.18 (PANAMAX), 14.2, 79.9, 0.15 (SUEZMAX), 13.3, 74.4, 0.14 (VLCC)

- Additional weights may arise due to various strengthenings for specific operating conditions of the ship, such as:
  - Navigation in ice; for example, an ordinary cargo ship may need additional steel weight strengthenings of +40 to 50 t, as specified by the classification societies' rules
  - For lifting of heavy weights, local strengthening of up to +80 t
  - Owner's specific additional requirements up to 2~3% W_{ST}.

The number of decks and bulkheads, if deviating from 'normal' practice, affects also the steel weight.

Effect of speed: A high Froude number requires slender hull form (high slenderness ratio) and consequently causes an increased W_{ST}/Δ and also a change of W_{ST}/(LBD). While the corresponding decrease of the block coefficient C_B causes an increase of the ratio W_{ST}/Δ, generally the ratio W_{ST}/(LBD) may decrease, if the main dimensions remain constant, which means that the reference displacement is reduced, as well as the transport capacity of the ship (see Table 2.13). For keeping the same transport capacity, the dimensions would need to be changed, thus the weight will be finally increased. In addition, an increase of the Froude number implies an increased machinery weight and generally increased values of weight coefficients W_L/Δ and $W_L/L\,B\,D$ (see Table 2.20).

Effect of the main dimensions: An increase of the absolute size of the ship, namely, as expressed by the increase of the product $L{\cdot}B{\cdot}D$ and a reduction of L/D or C_B, affect with decreasing trend the coefficients W_{ST}/(LBD) (see Table 2.21).

a3. *Effects on the weight of accommodation and outfitting*:

Determinant factors regarding the values of the corresponding coefficients are the followings

- Number of passengers and crew
- Accommodation quality
- Type and number of loading/unloading equipment
- Extent of reefer cargo spaces, if any
- Extent of insulation works etc.

Table 2.21 Effect of main dimensions on the weight of steel structure and outfitting. (Strohbusch 1971)

Bulk-carrier					
LBD [m^3]		110,000		200,000	
C_B		0.85	0.75	0.85	0.75
W_{ST}/LBD [kp/m^3]	L/D = 14	116	108	113	106
	L/D = 13	111	104	109	103
W_{OT}/LBD [kp/m^3]		17		13	

Comments made for tankers in footnote to Table 2.20 hold also herein. Characteristic values for modern large size bulkcarriers: $W_{ST}/L\,B\,D \approx 76.1$ [kp/m^3] for $L\,B\,D \approx 282{,}000$ m^3

In general, the coefficients decrease with the increase of the absolute size of the ship (see Table 2.21).

a4. *Effects on the machinery weight*
The basic influencing factors as to the coefficients for the machinery weight are:

- Required speed and installed engine power
- Type of main engine (diesel low turning speed—medium speed—high speed, turbine) and transmission mode (with or without gearbox)
- The position of the engine room significantly affects the coefficient for the weight of shaft (and propeller) W_{MS},
- The type of ship and the required electric power for servicing auxiliary facilities significantly affect the W_{MR} coefficient (rest machinery), for example, passenger and reefer ships.

Indicative values for the ratio of the installed power of the main engine to the ship's displacement are shown in the following Table 2.22, with the following notes:

- MONOHULL-AQUASTRADA: Large (L > 100 m), high speed (V > 40 kn) mono-hull type RO-RO passenger ship built by the Italian shipyard RODRIQUEZ (1993)
- CATAMARAN: twin-hull type ship for medium (seldom slow) and high speeds (planning or semi-planning/semi-displacement mode) with hybrid development features
- SWATH (Small Waterplane Area Twin Hull): Hybrid type CATAMARAN with small waterplane area for (low), medium to relatively high speeds (up to 35 kn absolute speed, depending on ship size); characterized by excellent seakeeping performance, while sustaining high speed
- SES (Surface Effect Ship): Hybrid type CATAMARAN with air cushion support for high-speeds ($V > 40$ kn)
- WAVE PIERCER: Hybrid type CATAMARAN with very sharp entrance of the waterlines and wave-piercing protrusion at the bottom of the two hulls bridging deck in the bow region, for high speeds ($V > 35$ kn, depending on ship size; Table 2.23; Fig. 2.66)

Table 2.22 Ratios of installed propulsion power to displacement weight for various types of ships—synthesis by IHS Fairplay database (2011) and A. Papanikolaou (2002)

Ship type	P/Δ [PS/t]
Fast cargo ships (and containerships)	0.7–1.6
Slow cargo ships	0.4–0.6
Coaster cargo ships	0.4–0.6
Bulkcarriers	0.1–0.5
Tankers	0.10–0.35
Reefer ships	0.7–1.6
Fast passenger ships (non-high speed craft)	
Large	1.4–3.3
Small	1.6–3.3
Medium to slow passenger ships	
Large	1.1–1.2
Small	1.0–2.8
Tugboats (seagoing)	up to 6.0
Advanced Marine Vehicles (very high speed crafts)	
MONOHULL-AQUASTRADA	$\cong$36.5
CATAMARAN	$\cong$25.0
SWATH	$\cong$20.0
SES	$\cong$35.0
WAVE PIERCER	$\cong$26.0
HYDROFOIL	$\cong$63.0

Advanced Marine Vehicles (AMV): These are generally high speed ships and boats of unconventional design and high operational performance (see also the following graph by A. Papanikolaou for the route of developments)

Table 2.23 Comments on the development chart of Advanced Marine Vehicles (AMVs) (Papanikolaou 2002)

1. ACV: **A**ir **C**ushion **V**ehicle - Hovercraft, excellent calm water and acceptable seakeeping (limiting wave height), limited payload capacity.
2. ALH: **A**ir **L**ubricated **H**ull, various developed concepts and patents, see type STOLKRAFT.
3. Deep V: ships with *Deep V* sections of semi-displacement type acc. to E. Serter (USA) or of more planing type, excellent calm water and payload characteristics, acceptable to good seakeeping, various concepts AQUASTRADA (RODRIQUEZ, Italy), PEGASUS (FINCANTIERI, Italy), MESTRAL –ALHAMBRA (BAZAN, Spain), CORSAIR (LEROUX & LOTZ, France).
4. FOILCAT: Twin hull (**cat**amaran) hydro**foil** craft of KVAERNER (Norway), likewise MITSUBISHI (Japan), excellent seakeeping (but for limited wave height) and calm water characteristics, limited payload.
5. LWC: **L**ow **W**ash **C**atamaran, twin hull superslender semi-displacement catamaran with low wave-wash signature of FBM Marine Ltd. (United Kingdom), employed for river and closed harbour traffic.
6. LSBK: **L**ängs **S**tufen- **B**odenkanalboot- **K**onzept, optimized air-lubricated twin hull with stepped planing demihulls, separated by tunnel, aerodynamically generated cushion, patented in Germany.
7. MIDFOIL: Submerged Foil-body and surface piercing twin struts of NAVATEK-LOCKHEED (USA).

Table 2.23 (continued)

8. MONOSTAB: Semi-planing monohull with fully submerged, stabilizing stern fins of RODRIQUEZ (Italy).
9. MWATH: **M**edium **W**aterplane **A**rea **T**win **H**ull Ship, as type SWATH, however with larger waterplane area, increased payload capacity and reduced sensitivity to weight changes, worse seakeeping.
10. SES: **S**urface **E**ffect **S**hip, Air Cushion Catamaran Ship, similar to ACV type concept, however w/o side skirts, improved seakeeping and payload characteristics.
11. SLICE: Staggered quadruple demihulls with twin struts on each side, acc. to NAVATEK-LOCKHEED (USA.
12. SSTH: **S**uper**s**lender **T**win **H**ull, semi-displacement catamaran with very slender long demihulls of IHI shipyard (Japan), similar to type WAVEPIERCER.
13. STOLKRAFT: Optimized air-lubricated V-section shape catamaran, with central body, reduced frictional resistance characteristics, limited payload, questionable seakeeping in open seas, patented by STOLKRAFT (Australia)
14. Superslender Monohull with Outriggers: Long monohull with two small outriggers in the stern part, EUROEXPRESS concept of KVAERNER-MASA Yards (Finland), excellent calm water performance and payload characteristics, good seakeeping in head seas.
15. SWATH Hybrids: SWATH type bow section part and planing catamaran astern section (STENA's HSS of former Finyards, Finland, AUSTAL hybrids, Australia), derived from original type SWATH & MWATH concepts.
16. SWATH: **S**mall **W**aterplane **A**rea **T**win **H**ull Ship, synonym to SSC (**S**emi-**S**ubmerged **C**atamaran of MITSUI Ltd.), ships with excellent seakeeping characteristics, especially in short period seas, reduced payload capacity, appreciable calm water performance.
17. TRICAT: Twin hull semi-displacement catamaran with middle body above SWL of FBM Marine Ltd. (United Kingdom).
18. TRIMARAN: Long monohull with a pair of small outriggers, introduced by Prof. D. Andrews–UCL London (United Kingdom), tested as large prototype by the UK Royal Navy (TRITON), similarities to the Superslender Monohull with outriggers concept of KVAERNER-MASA; excellent calm water performance; problematic seakeeping in oblique and beam seas; concept later developed and as pentamaran (with two pairs of outriggers).
19. TSL-F - SWASH: Techno-Superliner Foil version developed in Japan by shipyard consortium, submerged monohull with foils and surface piercing struts.
20. V-CAT: Semi-displacement catamaran with V section shaped demihulls of NKK shipyard (Japan), as type WAVEPIERCER.
21. WAVEPIERCER: Semi-displacement catamaran of INCAT Ltd. (Australia), good seakeeping characteristics in long period seas (swells), good calm water performance and payload characteristics.
22. WEINBLUME: Displacement catamaran with staggered demihulls, introduced by Prof. H. Söding (IfS-Hamburg-Germany), very good wave resistance characteristics, acceptable seakeeping and payload, name to the honour of late Prof. G. Weinblum.
23. WFK: **W**ave **F**orming **K**eel High Speed Catamaran Craft, employment of stepped planing demihulls, like type LSBK, but additionally introduction of air to the planing surfaces to form lubricating film of micro-bubbles or sea foam with the effect of reduction of frictional resistance, patented by A. Jones (USA)
24. WIG: Wing In Ground Effect Craft, various developed concepts and patents, passenger/cargo carrying and naval ship applications, excellent calm water performance, limited payload capacity, limited operational wave height, most prominent representatives the ECRANOPLANS of former USSR.

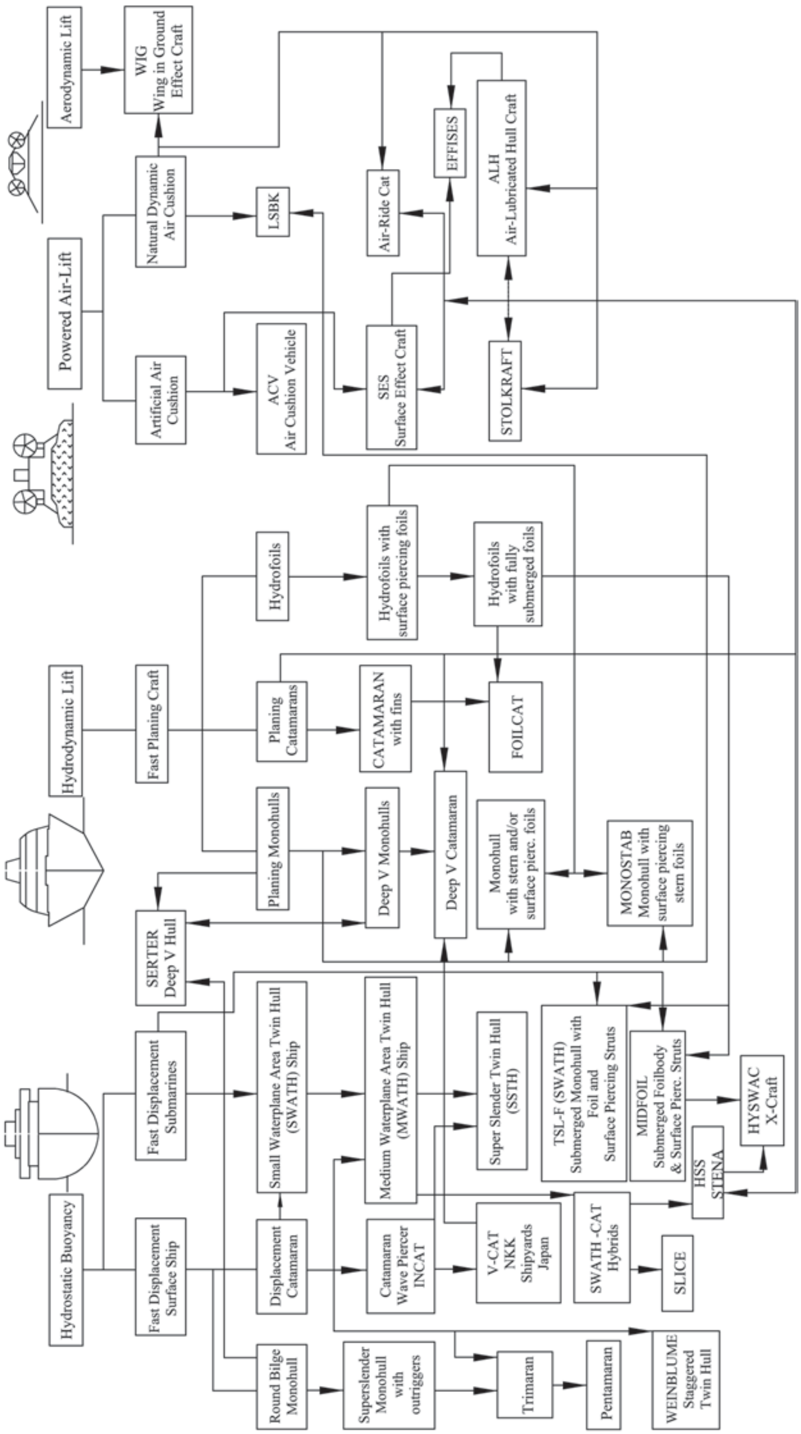


Fig. 2.66 Development of basic and hybrid types of advanced marine vehicles. (Papanikolaou 2002)

b. Specific effects on various types of ships

b1. *Cargo ships*: This paragraph applies only to cargo ships built under the provisions of the old tonnage/capacity regulations, distinguishing between “open” or “closed” type tonnage measurement (see Antoniou and Perras 1984). For the conversion of a cargo ship of open-type tonnage measurement to a corresponding ship of closed-type and for the same principal dimensions, the weight W_{ST} would increase by about 8 %, the W_{M} by 10 % and the displacement by 16 %, as well as the draught. It is estimated that with this conversion the transportation capacity may increase by about 20 %. In conclusion, a ship of “closed type” prevails in terms of weight distribution and exploitation (DWT) in comparison to an equivalent of “open type” measurement. However, in the new international tonnage regulations the distinction between “open” or “closed” type measurement has been removed and a consistent way of measurement of the ship’s enclosed volume and tonnage came into force. Essentially, ships measured with the new international tonnage regulations correspond to ships of former “closed” type in terms of weight distribution and exploitation of capacity.

b2. *Tankers, Bulk carriers*: Generally, the weight W_{ST} relatively decreases with the increase of absolute size. However, due to limitation of drafts (what means increased beam and may be increased length), this trend can reverse for very large ship sizes.

b3. *Reefer ships*: They are distinguished by their relatively high steel weight due to the slenderness of the hull form; they also have relatively high machinery weight due to the relatively high speed (large installed engine power); also relatively high outfitting weight, due to the weight of reefer facilities/outfitting (including increased electric energy consumption). In conclusion it shows a relatively large light ship weight and low ratio DWT/Δ (deadweight to ship displacement).

b4. *RoPax/Ro-Ro ferries:* Basically the same comments, as to the reefer ships, apply also to RoPax ships, though the reasons are partly different: their increased weights in the outfitting weight category are due to the large extent of accommodation spaces, the increased need for electrical energy (lighting, air-conditioning, etc.) and Ro-Ro loading outfitting (ramps etc., if not counted in the steel weight). Hence, they also dispose high lightship weight and small ratio DWT/Δ (classical *volume carrier*).

The above comments are expressed quantitatively with the shown typical values of weight coefficients in Table 2.1 (Sect. 2.1).

2.15.4 Structural Weight

As defined in Sect. 2.15.1, the weight of the ship’s structure W_{ST} includes the steel weight of the main hull, of the superstructures (even if party of wholly not made from steel, for example, light weight superstructures from aluminum alloys), as well as of some heavy steel fittings (like masts or derricks, etc.), which could be as well have been included in the W_{OT}.

A. **Simplified methods for W_{ST} calculation (preliminary design phase)**

A1. *Method of Harvald and Jensen (1992)* (see Friis et al. 2002)

The method is based on structural weight data of ships built in Danish shipyards; the data involve a large number of ships built in the decade of 80ties and until the early 90ties. The method uses as a basis the approximate enclosed volume of steel structure V_C, which includes the volume of the main hull, of the superstructures and deckhouses; furthermore, a coefficient for the steel structural density C_S is employed.

We assume

$$V_C \approx LBD + \text{Volume of superstructures and deckhouses}$$

and

$$W_D \equiv \text{DWT}, C_S = W_S / V_C, W_S \equiv W_{ST}$$

We may use the following diagrams, in which the steel structural coefficient C_S is given as a function of displacement Δ (Fig. 2.67), of W_D (≡DWT) (Fig. 2.68) and the enclosed volume of V_C (Fig. 2.69).

The curves in Fig. 2.67 can be mathematically expressed also by the relationship:

$$C_S(\Delta) = C_{S0} + 0.064\ \exp(-0.5\log_{10}\Delta + 1 - -0.1(\log_{10}\Delta - 2)^{2.45}) \qquad (2.114)$$

The C_{S0} for various types of ships is given in the following table (Table 2.24; Figs. 2.68 and 2.69).

From the analysis of data (regression fitting), the following approximate relationships are obtained, expressing the DWT and the enclosed volume V_C as a function of displacement Δ.

Cargo ships and bulk carriers

$W_D = 0.1951 \bullet \Delta^{1.13}$

$V_C = 12.127 \bullet \Delta^{0.883}$

Tankers

$W_D = 0.4464 \bullet \Delta^{1.05}$

$V_C = 4.674 \bullet \Delta^{0.915}$

Rail ferries

$W_D = 0.00363 \bullet \Delta^{1.5}$

$V_C = 1.951 \bullet \Delta^{1.12}$

A2. *Method of Cubic Number Coefficient CNC*

Assumption The W_{ST} weight varies proportionally to the product of the main dimensions $L \cdot B \cdot D$, expressing approximately the enclosed volume of the ship's structures:

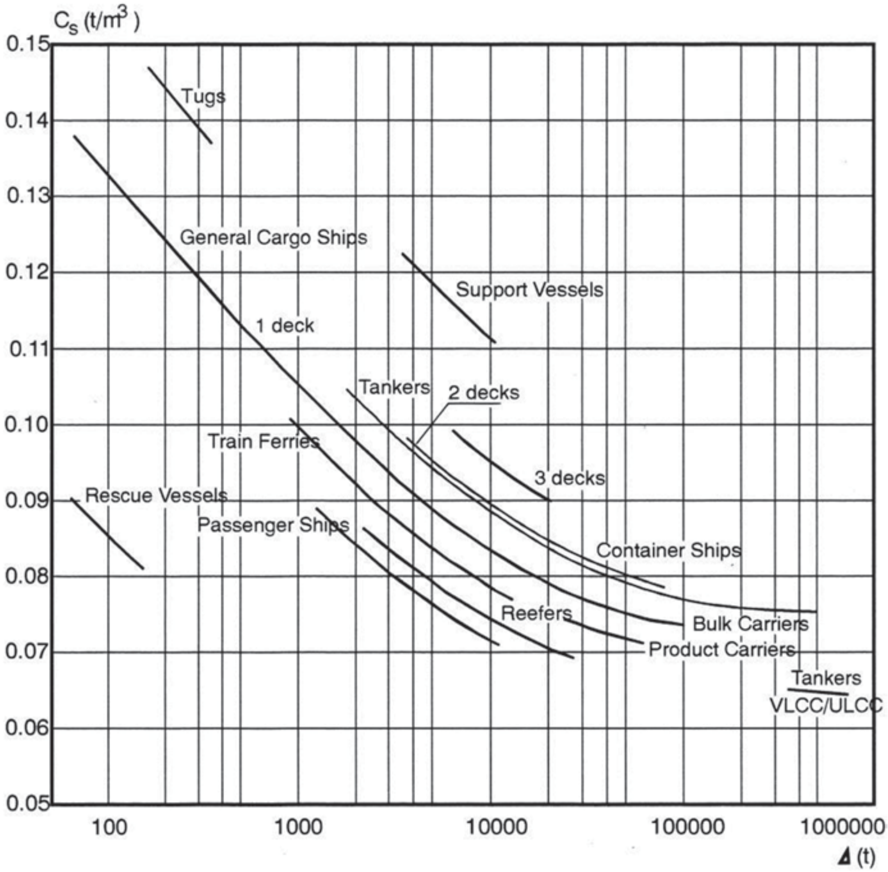


Fig. 2.67 Steel structural weight coefficient C_S versus displacement Δ by Harvald and Jensen. (Friis et al. 2002)

$$CNC = \frac{W_{ST}}{LBD}$$

Application Given the W_{ST}, L, B, D of a *parent*, geometrically similar, ship (index 0), it is assumed for the under design ship (index 1):

$$(W_{ST})_1 = (\mathrm{CNC})_0 \cdot L_1 \cdot \mathrm{B}_1 \cdot D_1$$

Corrections For differences of the ship's main characteristics fr om those of the parent ship, the cubic coefficient of CNC can be corrected as following:

$$\mathrm{CNC} = (\mathrm{CNC})_0 \cdot \mathrm{K}_1 \cdot \mathrm{K}_2 \ldots \mathrm{K}_n$$

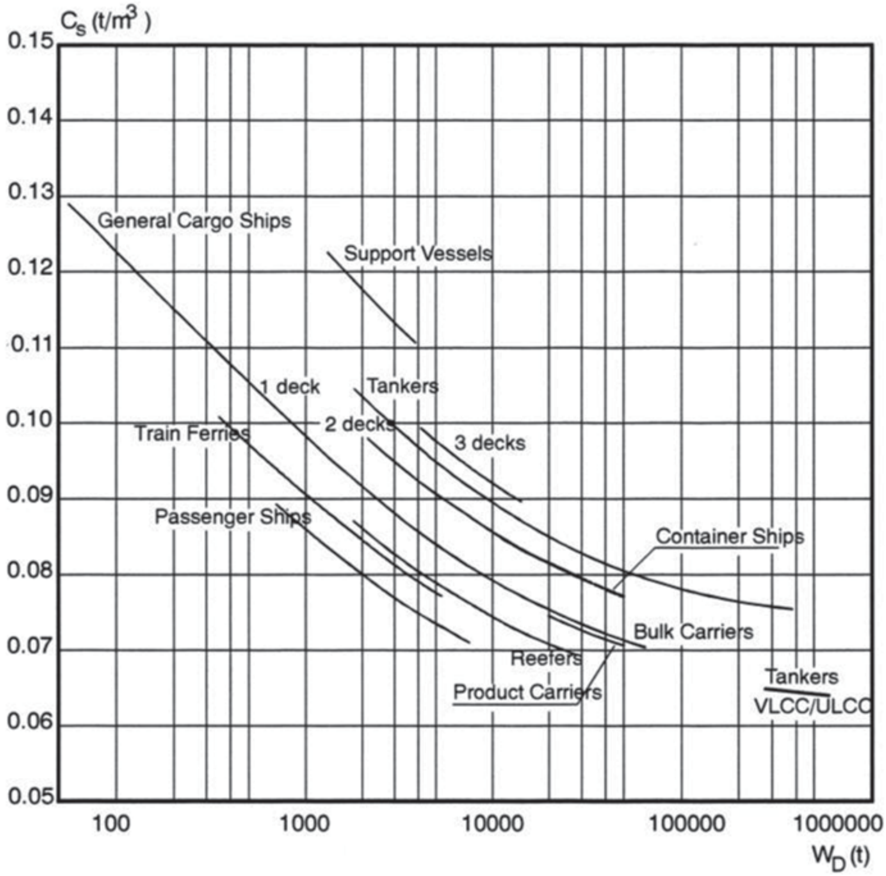


Fig. 2.68 Steel weight coefficient C_S versus the DWT by Harvald and Jensen. (Friis et al. 2002)

1. Correction for different C_B:

$$K_1 = (1 + 0.5C_B)_1 / (1 + 0.5C_B)_0$$

2. Correction for different L/D:

$$K_2 = (L/D)_1/(L/D)_0.$$

Comments

1. The method is simple and satisfactory, if there are sufficient data from similar ships available.
2. The accuracy of the method is sufficient for the initial design stage.

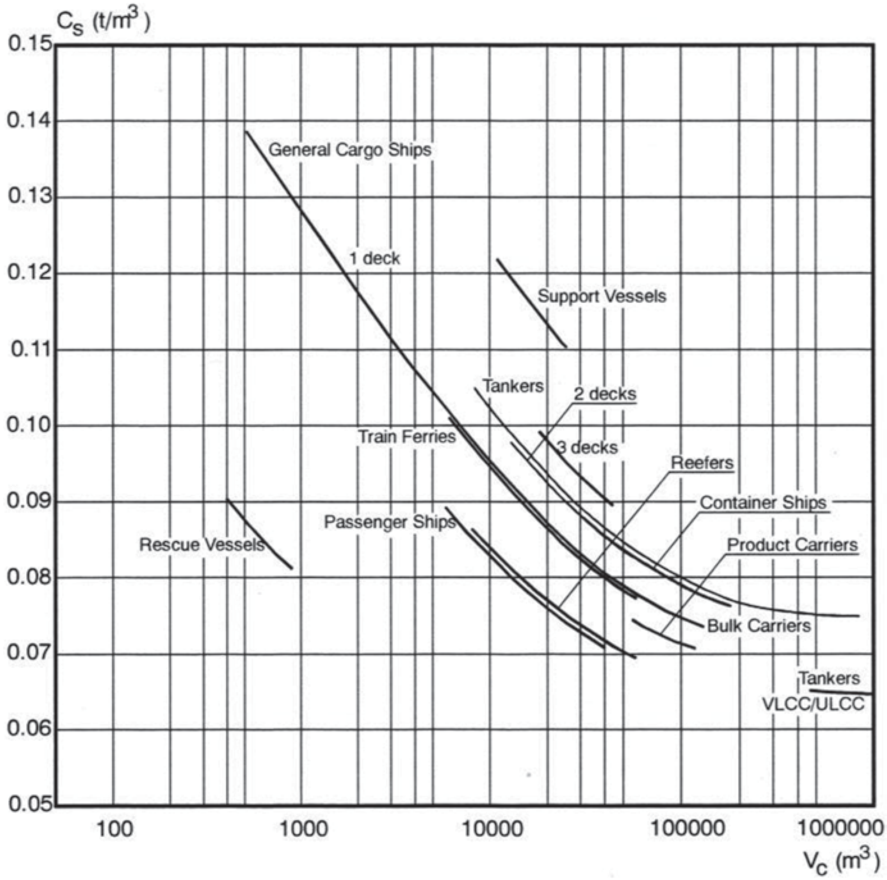


Fig. 2.69 Steel structural weight coefficient C_S versus the enclosed volume V_C by Harvald and Jensen. (Friis et al. 2002)

Table 2.24 C_{S0} for various types of ships

Ship type	C_{S0} (t/m^3)
Support vessels	0.0974
Tugs	0.0892
Cargo ships (3 decks)	0.0820
Cargo ships (2 decks)	0.0760
Cargo ships (1 deck)	0.0700
Tankers	0.0752
Bulk carries	0.0700
Product carriers	0.0664
Train ferries	0.0650
VLCC	0.0645
Reefers	0.0609
Passenger ship	0.0580
Rescue vessels	0.0232

A3. *Difference Method*

Assumption The W_{ST} weight results from the corresponding weight of a *paren*t ship; individual differences of the main dimensions, of hull coefficients and of local structural strengthenings are taken into account as following:

$$(W_{\mathrm{ST}})_1 = (W_{\mathrm{ST}})_0 \cdot (1 + C_1 + C_2 + \ldots + C_6) \cdot (1 + C_7)$$

Corrections-Coefficients

Correction for different length, $\delta L = L_1 - L_0$	$C_1 = 1.0\ \delta L/L_0$
Correction for different breadth, $\delta B = B_1 - B_0$	$C_2 = 0.7\ \delta B/B_0$
Correction for different side depth, $\delta D = D_1 - D_0$	$C_3 = 0.4\ \delta D/D_0$
Correction for local strengthening components as to the length	$C_4 = 0.45\ C_1$
Correction for local strengthening components as to the breadth	$C_5 = 0.35\ C_2$
Correction for local strengthening components as to the side depth	$C_6 = 0.65\ C_3$
Correction for different C_{B}, $\delta C_{\mathrm{B}} = C_{\mathrm{B1}} - C_{\mathrm{B2}}$	$C_7 = 0.3\ \delta C_{\mathrm{B}}$

Comments

1. All correction coefficients C_{i} can be positive or negative according to the sign of the differences δL, δB, δD and δC_{B} (increase of decrease of relevant dimensions).
2. The method is easy to use and generally well applicable in the initial design phase, assuming the availability of satisfactory parent ship data.
3. The method proved very effective in computer-aided optimization procedures of the ship's initial design, in which the ship's main dimensions are varied parametrically.
4. The following effects are not included: effect of differences in the draft, in the extent of superstructures, and in the number of decks (as applicable).

A4. *Watson's Method* (Watson and Gilfillan 1976)

Assumption The W_{ST} weight can be calculated based on the equipment index/numeral E_{N} (**E**quipment **N**umerical) of the ship as defined by Lloyd's Register (LR):

$$E_N = L(B+T) + 0.8L(D-T) + 0.85\sum_{i=1}^{N1} h_{1i} l_{1i} + 0.75\sum_{i=1}^{N2} h_{2i} l_{2i}$$

where

N_1, h_{1i}, l_{1i}: number, height and length of deckhouses[20]
N_2, h_{2i}, l_{2i}: number, height and length of the superstructures[21]

[20] By definition, the breadth of deckhouses can be *up to* 0.92 B.

[21] The breadth of superstructures *is larger* than 0.92 B according to the provisions of the International Tonnage Measurement regulation.

Application Through Fig. 2.70, where the W_{ST} is presented as a function of E_N, the corresponding weight for a standard block coefficient C_B*, at the height 0.8D, equal to 0.70, can be calculated:

$$(W_{ST})^* = f(E_N), \text{Fig. } 2.70$$

Correction For the ship's C_B* (0.8D) ≠ 0.7, the following correction applies:

$$(W_{ST}) = (W_{ST})^* \cdot (1 + 0.05(C_B^* - 0.7))$$

where the coefficient C_{B1}*(0.8D) can be approximated through the value of $C_{B1}(T=D)$

$$C_{B1}^* = C_{B1} + (1 - C_{B1})(0.8D - \mathrm{T}) / 3\mathrm{T}$$

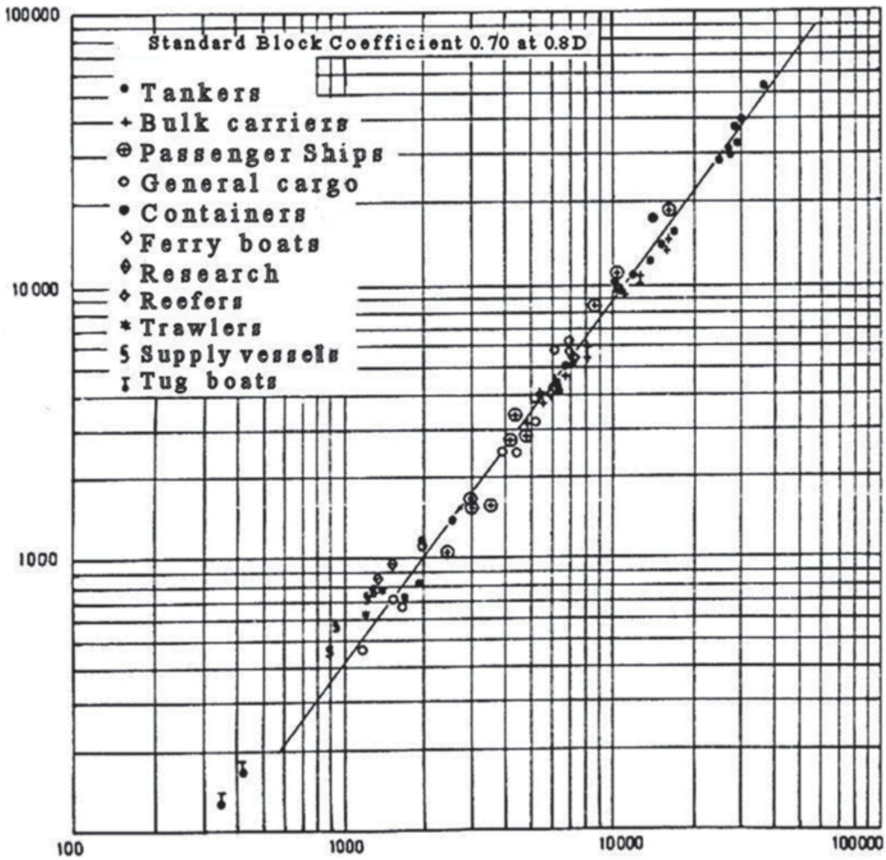


Fig. 2.70 Steel weight W_{ST} versus outfitting index E_N by Watson. (Watson and Gilfillan 1976)

Table 2.25 Steel weight coefficient by Watson (1998)

Ship type	Average value K	Fluctuation K [±]	Lower limit E_N	Upper limit E_N
Crude oil tankers	0.032	0.003	1,500	40,000
Chemical tankers	0.036	0.001	1,900	2,500
Bulkcarriers	0.031	0.002	3,000	15,000
Containerships	0.036	0.003	6,000	13,000
General cargo	0.033	0.004	2,000	7,000
Reefers	0.034	0.002	4,000	6,000
Coasters cargo	0.030	0.002	1,000	2,000
Offshore supply vessels	0.045	0.005	800	1,300
Tugs	0.044	0.002	350	450
Trawlers	0.041	0.001	250	1,300
Hydrographic vessels	0.045	0.002	1,350	1,500
RoPax	0.031	0.006	2,000	5,000
Passenger ships	0.038	0.001	5,000	15,000
Frigates/corvettes	0.023			

The above coefficients refer to structures built from 100 % mild shipbuilding steel. Given that a series of ship types today are built to some extent from higher tensile steel, the resulting weights by use of the above coefficients are expected to be slightly higher than today's standards (e.g., for tankers, bulkcarriers, containerships)

Comments

1. The method is simple and generally applicable in the initial design phase.
2. Due to its simplicity some basic ship features are neglected, which however may significantly influence the final estimation of the steel weight; for example, particularities of some ship types, number of decks and bulkheads etc.
3. The method has been improved by more recent studies of Watson (1998), namely:

$(W_{ST})^* = KE_N^{1.36}$, where K is listed in Table 2.25.

A5. *Danckwardt's Method* (Danckwardt 1961, Journal Schiffbautechnik)

Assumption The weight W_{ST} can be calculated as a function of the required volume of cargo spaces ∇_C, which includes the grain hold volume, the net volume of refrigerated cargo spaces (inside of insulation) multiplied by 1.3 ~ 1.5 (corresponding to the grain volume of refrigerated spaces) and finally, the volume of tanks *outside the* engine room and double bottom and between the forward and aft collision bulkheads.

The ratio W_{ST}/DWT is given as a function of DWT for various ∇_C/DWT values (see Fig. 2.71) for cargo ships up to DWT = 18,000 t. The curves are valid for "ordinary/standard" cargo ships with two decks and for a number of watertight bulkheads in conformity with standard classification societies' rules; the ship is assumed to be a cargo ship without passengers, without any special strengthenings and fully welded. The installed power of the propulsion plant is assumed to correspond to about 0.7 [HP] per [ton] DWT.

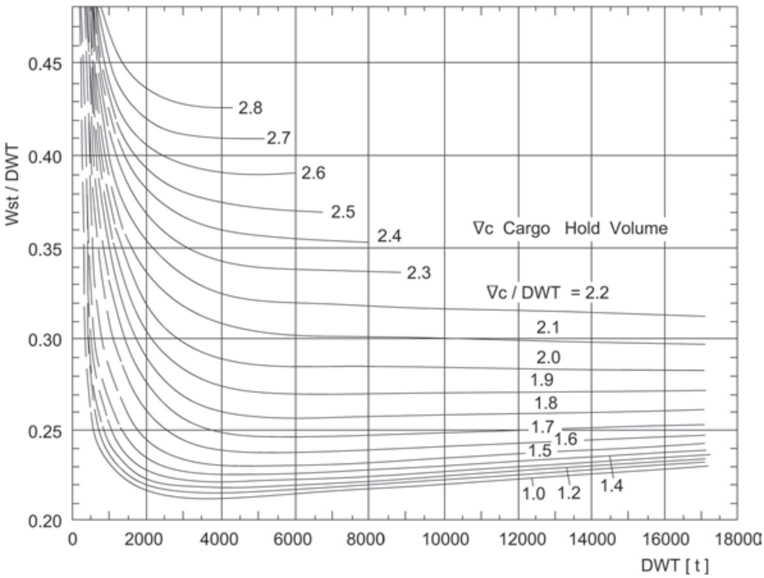


Fig. 2.71 Steel weight W_{ST} versus the DWT and volume ∇_C for dry cargo ships by Danckwardt. (Henschke 1964)

Corrections

1. Number of watertight bulkheads different from the regulations of classification societies, weight increase δW_{ST}/DWT:

$$+\text{One bulkhead:} \quad 0.25\%\text{DWT}$$
$$+\text{Two bulkheads:} \quad 0.31\%\text{DWT}$$
$$+\text{Three bulkheads:} \quad 0.50\%\text{DWT}$$

2. Strengthening for navigation in ice:

$$+2 \text{ to} + 9\%\text{DWT}$$

3. Strengthening for transportation of heavy bulk cargoes (ores)

$$\text{up to} + 6\%\,\text{DWT}$$

4. Strengthening for equipment of heavy lift derricks/cranes:

$$\text{up to} + 4\%\,\text{DWT}$$

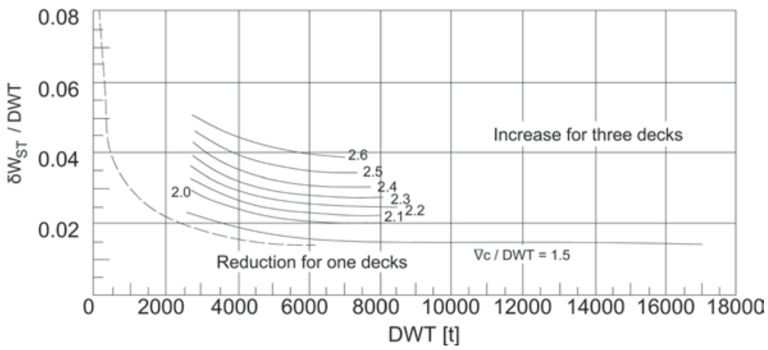


Fig. 2.72 Correction of steel weight by Danckwardt for number of decks different from the standard. (Henschke 1964)

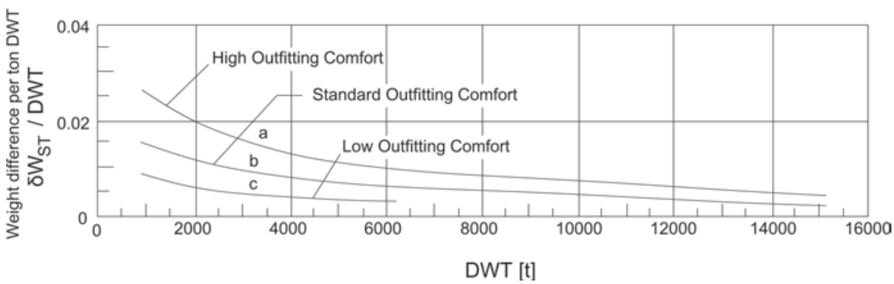


Fig. 2.73 Correction of steel weight by Danckwardt for quality of accommodation different from the standard (valid for up to 12 passengers). (Henschke 1964)

5. Number of decks different from the standard two (2): correction in accordance with Fig. 2.72.
6. Number of passengers up to 12: correction in accordance with Fig. 2.73.
7. Correction for the size of engine room different from the standard, which corresponds to P/DWT = 0.7 HP/ton, according to Fig. 2.74.

Note:

(1) This method is mainly applied to general cargo ships, with good results, though basic data of method are outdated.
(2) The reported corrections can be used in combination with other simplified methods, if the corresponding under assessment structural component of the parent ship is common.

A6. *General comments on the simplified methods for W_{ST} calculation (preliminary design phase)*

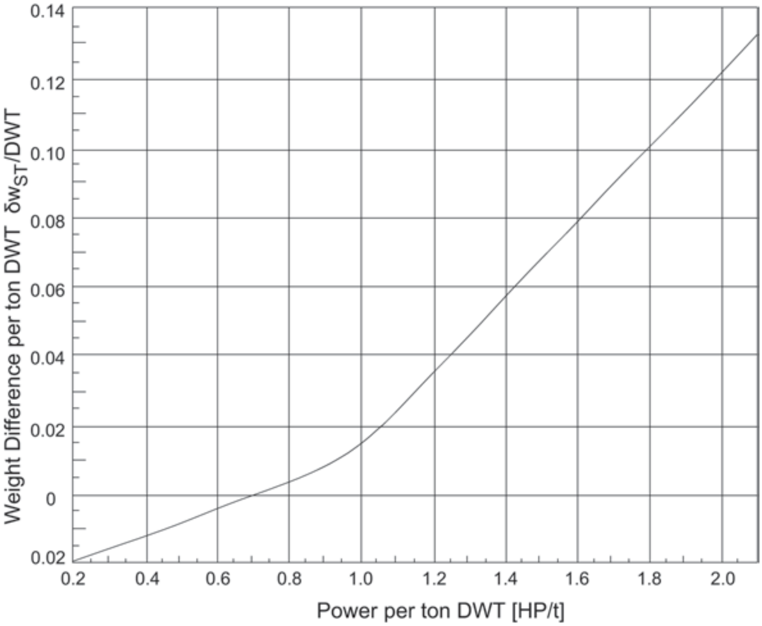


Fig. 2.74 Correction of steel weight by Danckwardt for main engine power different from the standard. (Henschke 1964)

It is considered that the accuracy of the approximation of W_{ST} through the above simplified methods is in the range of ± 5 %, but in practice for ships with special features the difference may be up to 8 ± 10 %. Such special conditions are for example:

- Differences in the requirements of various regulations (classification societies, national and international organizations).
- *Effect of new regulations*, for instance, the requirements of MARPOL for tankers concerning the use of segregated ballast tanks directly led to an increase of the number of tanks and consequently of the steel weight. Furthermore we have seen in recent years an increase of the steel weight of tankers with the implementation of OPA 90 and the revised MARPOL regulation (introduction of double-hull/skin tankers).
- *Effect of technological developments*: the steel weights generally decreased in recent years (though one needs to consider the counteracting weight increase due to the continuous introduction of new, more stringent safety regulations), for all types of ships, in view of improved methods for calculating the ship's strength (e.g., finite element methods) and optimizing the ship's structure for least weight; also, in view of the use of alternative materials other than the common mild shipbuilding steel, at least in some parts of the structure (higher-tensile steel for the strength deck and double bottom of tankers, bulkcarriers,

containerships, etc; aluminum alloys in the superstructures of passenger ships). Thus, comparing the steel weights of ships built during the 60s and 70s (for the same transportation capacity) with the contemporary ones, the values are actually today reduced, despite the weight increase due to the introduction of double skin hulls for tankers, or due to the more recent introduction of the Common Structural Rules of IACS class societies for tankers and bulkcarriers.

B. **More advanced methods of W_{ST} calculation (preliminary design stage)**

B1. *Strohbusch's Method* (Tech. University Berlin, 1928)

Feature Generalized method of relatively high accuracy, assuming that the structural plans of characteristic sections of a parent hull (or of the actual ship) are available.

Application

1. Calculation of the steel structural weight per meter of ship length for a limited number of characteristic sections of the ship.
2. Graphical representation of the curve $dW_{ST}/dx = w_{ST}(x)$ over the ship's length (see Fig. 2.75).
3. Calculation of the area under the curve, which corresponds to W_{ST}.
4. Addition of individual weights that are not taken into account in the weight per meter of length calculation of w_{ST} [ton/m].

$$W_{ST} = \int_{(L)} \frac{dW_{ST}}{dx} dx = \int_{(L)} w_{ST}(x) dx \cong \sum_{(N)} w_{ST}(x_i) \delta x_i$$

B2. *Vollbrecht–Többicke's Method* (1937–1948)

Feature Generalized method of satisfactory accuracy, if there are data from similar ships available.

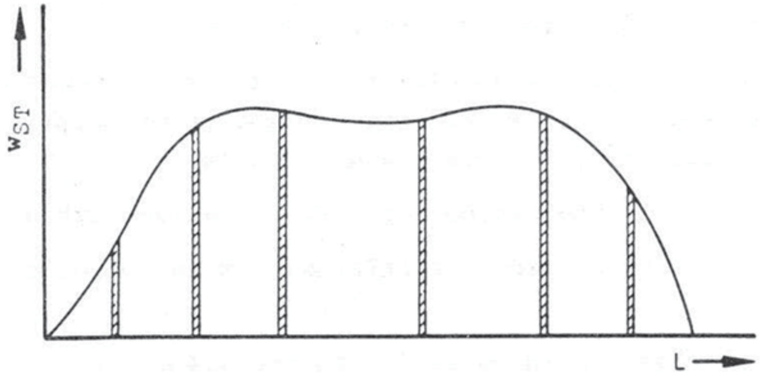


Fig. 2.75 Steel weight calculation by the method of Strohbusch

Application

1. Calculation of the steel weight for 1 m length of the *midship* section (similar to the method Strohbusch): (w_{ST}) [ton/m]
2. Calculation of W_{ST} for the ship based on the relationship:

$$W_{ST} = (w_{ST}) \, L \cdot C$$

where the constant C depends on the ship type, the ship's block coefficient and any special/unique features of the ship under design. This method can easily be adapted to various types of ships, if there are available data of parent ships for the approximation of C.

B3. *Schneekluth's Method* (Tech. Hochchule Aachen, 1967) (Schneekluth 1985)

Feature Synthetic method of good accuracy especially for dry-cargo ships (originally the method was developed for such ships); however, it is possible to apply it also to other ship types (e.g., tankers). It *does not include* the weight of *superstructures*, which can be calculated by the method of Müller–Köster (see Sect. 2.15.4, B4).

Assumptions (Original Method)

1. Dry cargo ships with continuous deck and bulkheads extending to the same deck
2. Constructional elements, for example, plate thickness, number of bulkheads, height of double bottom, in according to the Germanischer Lloyd Classification Society, Regulations of 1967, Class 100 A4
3. Hull form of the ship without a bulbous bow or rudder heel
4. Single-screw ships driven by diesel engines and with the engine room abaft
5. Breadth of hatchways approximately 0.4B + 1.6 m and overall length of hatchways approximately 0.5 L
6. Included components of the steel structure:
   - High tanks in the engine room
   - Strengthening/coamings of hatchways
   - Engine casing construction
   - Bulwark of a length of 0.9 L
   - Chain locker, chain pipe, strengthening of anchor winch
   - Rudder bearings and shaft tube
7. The weight coefficients C_{ST}, given below, were increased by 10 % to account for the following elements that are not calculated individually:
   - Increased plate thickness (margin against corrosion)
   - Local reinforcements
   - Heavier construction beyond regulations
   - Main engine foundation/bearings, masts, derricks, rudder body
8. The following weights are not included:
   - Hatch covers
   - Specific reinforcements for high speed and high propulsive power

- Special constructions (e.g., high tanks beyond the standard in the engine room)
- Superstructures and deckhouses (see later on Müller-Köster's method, Sect. 2.15.4, B4)

Required data for the application

L[m]: length between perpendiculars ($\equiv L_{pp}$)
B[m]: breadth
T[m]: design draft
D[m]: side depth of the uppermost continuous deck
C_B[−]: block coefficient at design waterline (draft T)
C_{BD}[−]: block coefficient at height D
C_M[−]: midship section coefficient
S_F[m]: sheer height at FP
S_A[m]: sheer height at AP
b[m]: camber height at the midship section
n[−]: number of decks
∇_U[m^3]: volume below the uppermost continuous deck

If not known at the early design stage, the volume ∇_U can be approximated with the following formula:

$$\nabla_U = \nabla_D + \nabla_S + \nabla_b + \nabla_H \tag{2.115}$$

where

$$\nabla_D = L \cdot B \cdot D \cdot C_{BD} \text{(volume up to } D\text{)}$$

with

$$C_{BD} = C_B(T) + C_1(D - T)/T(1 - C_B)$$

and

$$\begin{aligned} C_1 &\cong 0.25 \text{ for ships with sections of small flare above waterline} \\ &\cong 0.40 - 0.7 \text{ for ships with significant sectional flare} \end{aligned}$$

Furthermore,

$$\nabla_S = L_S \cdot B \cdot (S_F + S_A) \cdot C_2 \text{(increase of volume due to sheer)} \tag{2.116}$$

with L_S: length of sheer extent ($\leq L_{PP}$) $C_2 = C_{BD}{}^{2/3}/6 \cong 1/7$

$$\nabla_b = L \cdot \mathrm{B} \cdot b \cdot C_3 \tag{2.117}$$

(increase of volume due to deck camber) with

$$C_3 \cong 0.7 \cdot C_{\mathrm{BD}}$$

and

$$\nabla_{\mathrm{H}} = \sum_{i}^{N} l_{\mathrm{Hi}} \cdot b_{\mathrm{Hi}} \cdot h_{\mathrm{Li}} \tag{2.118}$$

(increase of volume due to hatch coamings) with

l_{Hi}: length of hatch i
b_{Hi}: breadth of hatch i
h_{Li}: height of hatch/coaming i
N: number of hatches

Application The W'_{ST} *without* the weight of superstructures is given as a function of the estimated total volume ∇_{U} [m^3], of a coefficient of specific unit weight C'_{ST} [ton/m^3] and of various corrections:

$$\begin{aligned} W'_{\mathrm{ST}} = \nabla_{\mathrm{U}} C'_{\mathrm{ST}} \cdot & [1 + 0.033(L/D - 12)][1 + 0.06(n - D/D_0)] \cdot \\ & [1 + 0.05(1.85 - B/D)] \cdot [1 + 0.2(T/D - 0.85)] \cdot \\ & [0.92 + (1 - C_{\mathrm{BD}})^2] \cdot [1 + 0.75 C_{\mathrm{BD}} (C_{\mathrm{M}} - 0.98)] \end{aligned}$$

where $D_0 = 4$ m and $L/D \geq 9$.

The values of the coefficient C'_{ST}[ton/m^3] as a function of the ship type are:

Ship type	Length range
Normal cargo ship	60–180 m
$C'_{\mathrm{ST}} = 0.103[1 + 17(L - 110\mathrm{m})^2] \cdot 10^{-6}$	
Reefer ships	100–150 m
$C'_{\mathrm{ST}} = 0.106$ to 0.116	
Passenger ships	80–150 m
$C'_{\mathrm{ST}} = 0.113$ to 0.121	
Bulkcarriers	150–300 m
$C'_{\mathrm{ST}} = 0.108$ to 0.117	
Tankers	150–350 m
$C'_{\mathrm{ST}} = 0.112 + L\,[\mathrm{m}] \cdot 10^{-4} \cdot (0.95 \div 1.05)$	

While the original formula of Schneekluth was applied only to general dry cargo ships it was later on extended to other types of ships with relatively good success.

In general, the following applies:

1. For RoPax and ferry ships the use of the above relationship for passenger ships may be problematic, due to the significant reinforcement of decks for transporting heavy vehicles and the diversification of their structure.
2. For containerships a special relationship is given later on.

Corrections The weight of the ship's steel structure W_{ST}, calculated by the above formula, should be corrected as follows:

1. For transverse construction/strengthening system: +2.5 % W_{ST}
2. For the existence of bulbous bow: +0.4–0.7 % W_{ST} or consider the additional weight as a function of the bulb's volume: +0.4 t/m^3

Comments

1. The method was essentially developed following the approach of Strohbusch (see B1). The results from systematic calculations for different ships were synthesized in the above formula.
2. The advantages of this method are:
   - Relatively simple calculations with good results,
   - Can be easily coded in design computer programs,
   - Possible application to cargo ships with uncommon main dimensions and block coefficient
3. For the weight of superstructures, which is not included in the basic method, the method of Müller-Köster (see Sect. 2.15.4, B4) can be used
4. For calculating the steel structural weight of ships transporting standardized container (containerships), the above general formula shall be amended as follows:

$$W'_{ST} = \nabla_U \cdot C'_{ST} \cdot [1+0.002(L-120)^2] \cdot [1+0.057(L/D-12)] \cdot$$
$$[30/(D+14)]^{1/2} \cdot [1+0.1(B/D-2.1)^2] \cdot [1+0.2(T/D-0.85)] \cdot [0.92+(1-C_{BD})^2]$$

where

$$C'_{ST} = 0.090 \div 0.100, \text{average}: 0.093.$$

Constraints of Application (containerships)

L = 100–250 m
B = up to 32.25 m (Panamax)
L/B = 4.7–7.63 (small feeder ships: up to 4.0)
L/D = (8.12)–15.48 (lower limit of ship type: 10.0)
B/D = 1.47–2.38
B/T = 2.4–3.9 (for T = 0.61D)
= 1.84–2.98 (for T = 0.80D)
C_B = 0.52–0.716

(Extrapolation for small violations of the above limits is possible)

Table 2.26 Weights of container cell guides

Container		Weights of cell guides [t/TEU]	
Type	Length	Fixed	Detachable
Ordinary	20′	0.70	1.0
Ordinary	40′	0.45	0.7
Refrigerated	20′	0.75	–
Refrigerated	40′	0.48	–

Corrections (containerships):

1. For the exclusive use of a normal, mild shipbuilding steel (the formula applies to $L = 100$–180 m)

$$\delta W'_{\mathrm{ST}}[\%] = 3.5(L^{1/2} - 10) \cdot [1 + 0.1(L/D - 12)]$$

2. For trapezoidal midship section (containerships): generally reduction of W'_{ST}: $\delta W'_{\mathrm{ST}}[\%] \cong -5$
3. For raised double bottom beyond the regulations of Germanischer Lloyd classification society: for an increase of double bottom height by δh_{DB} and increase of double bottom volume by $\delta \nabla_{\mathrm{DB}}$ it shows:

$$(\delta W'_{\mathrm{ST}} / \delta \nabla_{\mathrm{DB}})(40 + 0.5\ \delta h_{\mathrm{DB}})10^{-3}[\mathrm{t/m^3}]$$

4. The weights of container cell guides *are commonly included* in W'_{ST}. Typical numbers of these weights are (Table 2.26):
5. The weights of the ducts of the cooling system (for reefer containers) and of the lashing equipment on deck are usually included in W_{OT} (see Sect. 2.15.5).

Center of weight W'_{ST}

In Schneekluth's method the approximation of the vertical position of mass center of W'_{ST} (without superstructures) is also included:

$$\overline{KG}'[\%D] = \left[44 + 0.155(0.85 - C_{BD}) \cdot \left(\frac{L}{D}\right)^2\right] \frac{D_S}{D}$$

where

$$(D_{\mathrm{S}}/D) = 1 + C_{\mathrm{BD}}^{2/3}(S_{\mathrm{F}} + S_{\mathrm{A}})/6D$$

(applies to ships with sheer extending up to at least amidships).

Corrections

1. For transverse framing-system of construction/strengthening: $-1\,\%$ D
2. For bulbous bow: $-0.4\,\%$ D
3. For L/B $\neq$ 6.5: $\pm 0.8\,\%$ D per $\delta(L/B) = \pm 1.0$

4. For $L \neq 120$ m: $+1\%$ D for $L = 60$ m and -1% D for $L = 180$ m

B4. *Weight of superstructures and deckhouses by Müller–Köster* (Müller–Köster 1973, Journal Hansa; Schneekluth 1985)

To calculate the total structural weight of the ship it is necessary to add the weight of superstructures and deckhouses to the main hull weight W'_{ST}, as calculated by Schneekluth.

Following Müller-Köster, this weight can be calculated as a function of the enclosed volume of the superstructures and in dependence on the location of the structural elements of superstructures and deckhouses.

Superstructures

According to the International Load Line Convention (ICLL), structures on the main deck (freeboard deck) with a distance of their side walls from the ship's side *less than*/equal to 4% B are assumed to be *superstructures* in the sense of ICLL. Such superstructures are:

a. **Forecastle:**

The volumetric weight (weight per volume unit) of a forecastle is:

$$C_{\text{FORECASTLE}} \cong 100\text{kp/m}^3 \text{for ship length } L \geq 140\,\text{m}$$
$$130 \text{ kp/m}^3 \text{for ship length } L \cong 120\,\text{m}.$$

Assumptions

$$\text{Height of forecastle}: h_{\text{FORECASTLE}} = 2.5 \text{ to } 3.25\text{m}$$
$$\text{Lengthof forecastle}: l_{\text{FORECASTLE}} = \text{up to } 0.2\ L_{\text{PP}}$$

Corrections

$$\delta C_{\text{FORECASTLE}}[\%] \text{up to} -10\%, \text{for} \quad l_{\text{BACK}} \cong 0.33 L_{\text{PP}}$$
$$\delta C_{\text{FORECASTLE}}[\%] \cong -5\% \text{ to} -10\%, \text{for } h_{\text{BACK}} > 3.25\,\text{m}.$$

b. **Poop**[22]:

$$C_{\text{POOP}}\, 75\text{kp} / \text{m}^3$$

[22] The poop deck is technically a raised *stern deck* that is rarely found on modern ships. In older sailing ships it could be seen as the elevated roof of the stern or "after" living quarters, also known as the "poop cabin". Also, with the helmsman at the stern, an elevated position was ideal for both navigation and observation of the crew and the sails. In modern history of shipbuilding, it could be seen until the 1960s on the "three island" type cargo ships, with the bridge and engine amidships (raised *quarterdeck*), and *forecastle* and *poop* decks at ship's ends. This concept was gradually displaced (and practically today disappeared) by the classical modern cargo ship arrangement, with the engine and bridge/superstructure placed astern, and having a 'flush' deck (extending unbroken from stem to stern, with no raised forecastle or quarterdeck) or keeping the forecastle at ship's bow region.

Assumption The poop extends up the forward bulkhead of the engine room, for engine room located abaft.

Corrections If the poop extends above a hold:

$$\delta C_{\mathrm{POOP}}[\%] \cong +20\%$$

Deckhouses

a. **Houses with living quarters:** Deckhouses extending over more than one deck are not considered as one single structure, but as consisting of several individual quarters, which are classified according to their vertical position above the main (uppermost continuous) deck. The weight of each quarter depends on its enclosed volume, but also on its structural density, which is clearly a function of the vertical position of the quarter and considers the loading of quarters located above the quarter in question. Quarters of superstructures, which are located directly on the main deck, are characterized as belonging to layer I (vertically extending up to Deck I), the ones above it to layer II, etc. (see sketch) (Fig. 2.76).

It is understood that if a deckhouse is located on the poop (or forecastle accordingly) then it begins with layer II.

The weight of the deckhouses depends on the following factors:

- Way of construction
- Length of ship
- Number of higher decks
- Height of decks
- Length of internal separating walls, if they are from steel/metal.
- Ratio of the upper deck (ceiling) area A_O, including the area of uncovered external *walkways*, to the actually covered (bottom) area of each deck A_U.

The following Table 2.27 gives the deckhouse weight per volume unit (structural density) as a function of the ratio A_O/A_U and layer position.

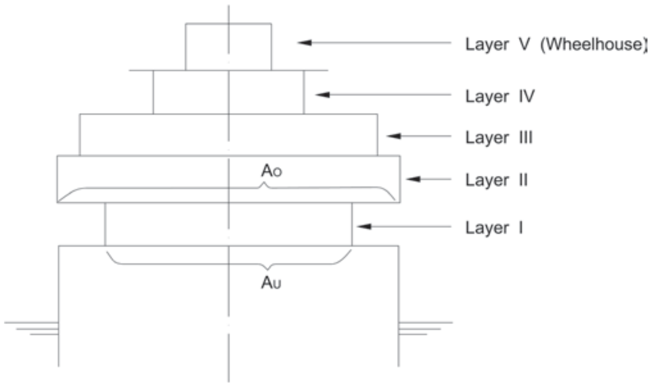


Fig. 2.76 Definition of individual layers for the calculation of the deckhouse weight by Müller–Köster

Table 2.27 Volumetric weight coefficients of deckhouses C_{DH} [kp/m^3] as a function of the position and A_O/A_U ratio according to Müller–Köster

Layer	I	II	III	IV	Wheelhouse
A_O/A_U					
1.0	57	55	52	53	40
1.25	64	63	59	60	45
1.5	71	70	65	66	50
1.75	78	77	72	73	55
2.0	86	84	78	80	60
2.25	93	91	85	86	65
2.5	100	98	91	93	70

The weight of a deckhouse section at the height/layer I to IV or at wheelhouse level is given by:

$$W_{DH} = C_{DH} \cdot A_m \cdot h \cdot k_1 \cdot k_2 \cdot k_3$$

where

C_{DH}[kp/m^3]: volumetric weight coefficient, given in Table 2.15; interpolation is possible for intermediate A_O/A_U values
A_m: mean area value: 0.5 (A_O+A_U)
h: height of deckhouse
$k_{1,}k_2,k_3$: corrections
k_1: correction for deckhouse height different from 2.6 m, namely $k_1=1+0.02\ (h-2.6\text{ m})$
k_2: correction for nonstandard length of internal walls (4.5 time of deckhouse section length)$=1+0.05(4.5-l_I/l_{DH})$, where 1_I: total length of internal walls, 1_{DH}: total length of deckhouse section
k_3: correction for ship length significantly different from $L_{PP}=150$ m, i.e., for $\delta L_{PP}>\pm 30$ m
$=0.95$ for $L_{PP}=100$ m $=1.10$ for $L_{PP}=230$ m
(interpolation for intermediate values possible).

The above relationships apply to *superstructures and deckhouses* with accommodation facilities regardless of their definition according to the ICLL regulations (for forecastle-poop, see previous references).

b. **Winch houses**: The volumetric weight coefficient of winch houses can be calculated by the following empirical formula:

$$C_{WH}\left[\text{kp}/\text{m}^3\right] = 48 + 4A_O / A_U\,(A_O/A_U + 8) + 18(150\text{m}^3 - \nabla_{WH})/\nabla_{WH}$$

where

$$\nabla_{WH} m^3 = A_U \cdot h_{WH}\,(\max: 150\text{m}^3)$$

Table 2.28 Correction factor for winch houses of derricks

Lifting capacity of derrick [t]	10	20	80	100	130	150
k_1	1.0	1.02	1.10	1.15	1.30	1.50

the volume of the winch house.

The winch house weight is given by:

$$W_{\mathrm{WH}} = C_{\mathrm{WH}} \cdot \nabla_{\mathrm{WH}} \cdot k_1$$

where

k_1: correction factor for winch houses of derricks with lifting capacity over 10 t, according to Table 2.28.

In case of very heavy lift derricks, which require special reinforcement of the foundations of the winch house, as well as of the winch basement, the above weights must be increased up to 70 % W_{WH}

The above formulas apply to the following values of $A_{\mathrm{O}}/A_{\mathrm{U}}$, h_{WH}, ∇_{WH}:

$$\begin{aligned} A_{\mathrm{O}} / A_{\mathrm{U}} &= 1.0 \div 3.0 \\ h_{\mathrm{WH}} &= 2.6 \div 3.2\ \mathrm{m} \\ \nabla_{\mathrm{WH}} &= 50 \text{ to } 200\mathrm{m}^3 \end{aligned}$$

When calculating the C_{WH}, the ∇_{WH} must not exceed 150 m^3, i.e., the value of the term in the last parenthesis of the formula should not be negative.

Weight centers of superstructures and deckhouses

For the vertical position of the weight centers, which are estimated as percentages of the height *h* of each deckhouse, and are calculated for deckhouses extending over more than one deck, for each section separately, it is assumed:

0.76–0.82 h, for deckhouses with internal walls
0.70 h, for deckhouses without walls

B5. *Other advanced methods*

a. **Steel structural weight by Puchstein (1961)** (Henschke 1964, Vol. 2, p. 457)

Application

"Standard" general cargo ships

Advantages

- High accuracy, but not for modern shipbuildings without the revision of individual coefficients and methods.
- Detailed breakdown of the weight of the steel structure into the weight of building blocks, which are approached separately (double bottom, shell plating, bulkheads, decks, strengthenings, superstructures and accommodation).

- The analysis of the steel structure into blocks facilitates the estimation of the centers of weight components.

Disadvantages

- Relatively tedious work
- It does not consider the longitudinal framing construction system.
- The individual elements of the method are to a great extent outdated; they can be updated/revised if there are available comparable data from similar ships.

Accuracy According to Puchstein: ±1 %

(only if data for modern ships are available).

Conclusions The obtained distribution of the steel weight of the individual components of the steel structure for the main ship hull (dry cargo ship) is very valuable:

Double bottom (includes the corresponding external shell)	25–35 % W_{ST}
External shell (includes sections/frames, without double bottom)	22–35 % W_{ST}
Bulkheads	4–8 % W_{ST}
Decks (includes deck strengthenings)	20–36 % W_{ST}
Other reinforcements (includes internal structures)	3–18 % W_{ST}

b. **Steel Structural Weight by Sturtzel (1952)** (Handbuch der Werften; 1959, Schiffahrts-Verlag Hansa, Hamburg)

Disadvantages Outdated data based on *riveted* shipbuildings; apply only indirectly to welded constructions.

c. **Steel Structural Weight by Röster-Krause (1929–1952)** (Henschke 1964, Vol. 1, p. 549)

Disadvantages Older data of Röster (1929) were revisited by Krause (1952); however, they do not correspond to modern constructions.

C. Analytical methods of calculating W_{ST}

C1. *Method of Blohm & Voss Shipyard by Carstens* (1967, Journal Hansa, Schiffahrtsverlag HANSA, Hamburg)

Features Generalized method of wide applicability, where W_{ST} is given as a function of the hull area and of the structural components.

Advantages

- High accuracy, wide applicability to different types of ships
- Detailed data on the effect of specific features of the construction, which can be used in combination with other methods:

Disadvantages

- Laborious work proportional to the targeted accuracy of the calculations

D. Weight of other components of the steel structure (Dudszus and Danckwardt 1982, Journal Schiffstechnik p. 243; Journal Hansa, 1975, Schiffahrtsverlag HANSA, Hamburg, p. 417):

Additional components of the steel structure, which must be taken into account in the calculations, except for a few methods that inherently include them (e.g., C1), are elaborated in the following.

1. **High fuel tanks:** Their weight is calculated based on the weight of their sidewalls (panel area), +30 % for strengthening.
2. **Additional bulkheads:** Their weight is obtained from the weight of the required plating, +40–60 % for the strengthening. For less bulkheads (with classification society's approval), we can reduce correspondingly the W_{ST}, which was estimated in advance.
3. **Strengthenings for heavy loads:** For heavy cargo loads in view of heavy bale cargo or ores special strengthening is required, especially of double bottom, according to the regulations of classification societies.
4. **Absence of planking of cargo hold floor:** Strengthening of cargo ships' holds' floor by 2 mm (according to GL), if planking overlay is missing; increase of strengthening by 5 mm or even more, if crab cranes or bulldozers are used for unloading.
5. **Height of double bottom:** If the double bottom height exceeds the standard size, for example, in Schneekluth's method the corresponding one specified by GL rules, an additional weight per unit volume difference of 100 kp/m^3 must be taken into account. Assumption: longitudinal frame strengthening except of at the ends of the ship, where transverse section framing prevails.

For the transverse framing construction system of double bottom, the volumetric unit weight is approximately:

$$C_{\mathrm{DB}}[\mathrm{kp/m^3}] \cong 100 + 0.5 \cdot h_{\mathrm{DB}} / (h_{\mathrm{DB}})_{\mathrm{NORM}} \text{ according to GL}$$

Assumption Floor plating on each section and lateral side girders every 4 m approximately. If the lateral side girders are fitted more densely, the coefficient C_{DB} must be increased by +30 %.

The volume of the double bottom can be approximated by[23]:

$$\nabla_{\mathrm{DB}}[\mathrm{m^3}] = L \cdot B \cdot h_{\mathrm{DB}} \cdot [C_{\mathrm{B}} - 0.4(T - h_{\mathrm{DB}})^2] / [T^{2 \cdot}(1 - C_{\mathrm{B}})^{0.5}]$$

where h_{DB}[m] the maximum height of double bottom.

[23] The *minimum double bottom height* for dry cargo and passenger ships, as specified in SOLAS, is B/20 or 2 m, whichever is less (*but not less than 760 mm*). For RoPax ships with large lower holds, this changes to B/10 and 3 m, whichever is less (SOLAS 2009). The minimum requirements for tankers are led down in MARPOL.

Table 2.29 Ice strengthening according to classification societies

	Ice classes					
Germanischer Lloyd	E	E1	E2	E3	E4	Icebreakers for navigation in North. and South Pole
Finish Lloyd		IC	IB	IA	IA Super	
$\delta W'_{ST}$ [%]	1–2	4	8	13	16	up to 180

6. **Engines' foundation:** For particularly powerful engines, especially heavy slow-speed diesel engines without gearbox, an enhanced strengthening of their foundation by approximately 3.6 kp/kW is required, or to be taken according to the formula:

$$\delta W_{ST}[t/kW] = 27/[(n+250)\cdot(15+P_B\cdot 10^{-3})]$$

where n [RRM]: number of engine revolutions per minute, P_B [kW]: engine break horsepower.

7. **Hatch coamings:** Continuous hatch coamings: ~0.090 t/m^3. Noncontinuous coamings: ~0.060 t/m^3: The values refer to the volume enclosed by the coamings of the hatchways above the deck.
8. **Reinforcements for corrosion:** If anticorrosion measures were considered appropriately, for example, the use of special coatings, the reinforcements of the plate thicknesses due to corrosion can be neglected, which leads to a reduction of W_{ST}. For a large tanker this can be: −3 to −5 % of the W_{ST} (main hull).
9. **Strengthening for navigation in ice** (Table 2.29)

E. Reduction of structural weight—Use of higher-tensile steel and aluminum alloys

In addition to the significant effect of the main dimensions, particularly of L and D, and form coefficients, particularly of C_B, on the steel/ship structural weight, the limited use of alternative materials or higher tensile steels in certain cases, next to the common shipbuilding steel (mild steel), can reduce the ship's total structural weight and has a positive effect on the position of the center of gravity of the hull structure.

E1. Use of higher-tensile steel

Higher tensile steels (HTS), with a yield strength (YS) of 315 to 355 MN/m^2 or [MPa] and ultimate tensile strength (UTS) of up to 620 [MPa], compared to the common (mild steel) shipbuilding steel (YS 235 to UTS 490), are used *locally* in merchant shipbuilding with special requirements on strength, for example, in the bottom/deck areas of large tankers VLCC and ULCC, bulkcarriers and containerships, as well as in structural blocks of large offshore structures. According to available data of actual constructions (Lamb eds. 2003), the proportion of higher tensile steel in large tankers is between 10 % and 38 % in extreme cases. It is estimated that using higher tensile steel locally on a tanker or a bulk-carrier (deck and bottom areas), the steel weight can be reduced by about 5~7 %. Certainly, higher tensile

steels, along with titanium alloys, constitute the main construction material for naval submarines and other warships.

The negative aspects and some attention points of using higher tensile steel are summarized in the following:

- As the modulus of elasticity of higher tensile steel does not change significantly in comparison to the corresponding one of mild steel, it is not possible to reduce the plate thicknesses directly proportional to the higher tensile strength, because loadings on compression stresses (buckling problems) remain roughly the same, thus it would lead to serious strength problems, if plating is strongly reduced. The buckling issues require additional thicknesses/reinforcements, resulting in a mitigation of the weight savings from using higher tensile steel.
- The *fatigue* strength of higher tensile steel is not significantly higher than that of the common mild steel.
- The corrosion of the plating over the years does not change significantly, thus practically the effect is more drastic since it leads to further reduction of an already reduced thickness of plating.
- There are surcharges on the construction cost, not only because of the increased material cost, but also due to the required extra effort in working hours for the welding.
- Finally, there were, in recent time, reports about problems regarding the quality of some newbuildings and conversions of large tankers and bulkcarries that were attributed to the quality of fitted HTS. Because a HTS construction is comparably more dependent on the quality of the fitted material, this is a very serious point of concern that needs to be carefully considered in the selection and quality control of the used steel material.

Some of above mentioned problems regarding the use of higher tensile steel, and generally regarding the sufficiency of strength of recent shipbuildings, led the classification societies of IACS (http://www.iacs.org.uk) to revise their regulations by introducing in year 2006 the Common Structural Rules (CSR) for the construction of tankers and bulkcarriers. These rules are in the direction of more rigorous construction and increased plating thicknesses. This was also in line with a proposal of the Greek delegation to IMO (together with Bahamas Islands) to consider the adoption of improved construction standards for new buildings (Goal Based Standards-GBS; Fig. 2.77).

E2. Use of light metals

Light weight materials, like aluminum, or better aluminum–magnesium alloys, are used for the construction of deckhouses and other individual structural components (e.g., funnels) of the ship's structure. Furthermore, they are the main construction material[24] for small vessels (up to $L \approx 40$ m) and high speed crafts in general.

[24] It should be noted that the largest ship ever built *entirely* from aluminum alloy was the high-speed hybrid SWATH catamaran "HSS1500" of STENA LINES, with LOA 126 m, beam 40 m and service speed 40 knots (Fig. 2.78).

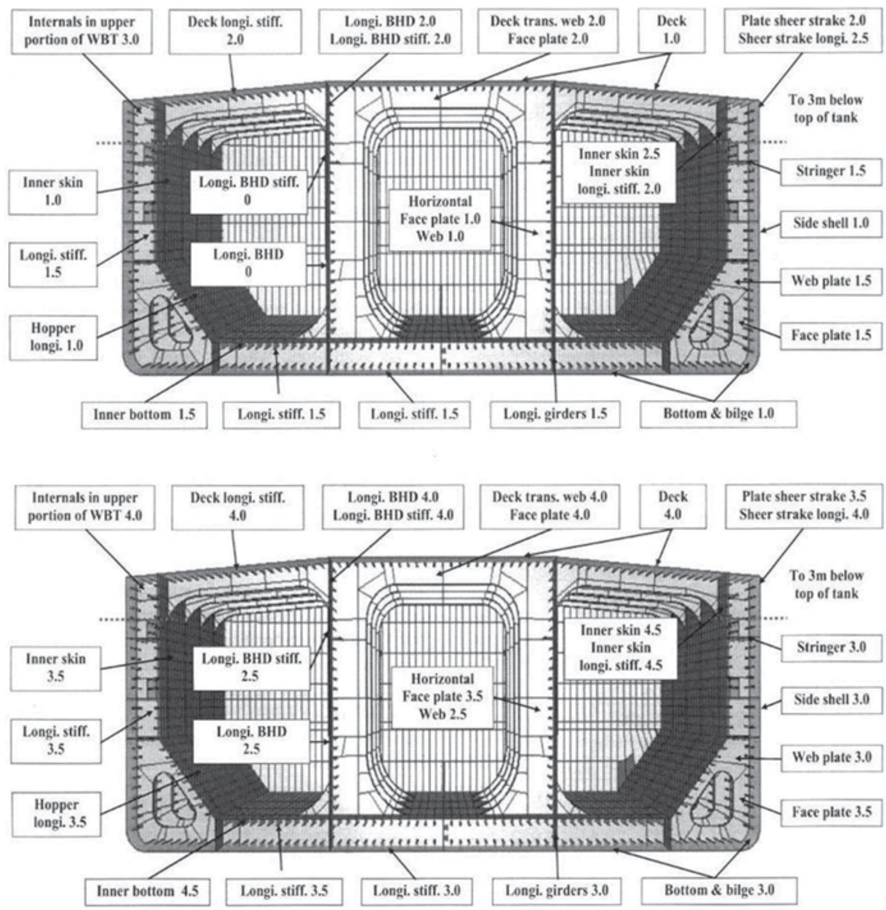


Fig. 2.77 Corrosion margins for tanker SUEZMAX (DWT: 158,000 t) according to old class society regulations (*upper figure*) and the new regulations (*bottom figure*) of IACS (Common Structural Rules). (Paik et al. 2009)

Compared to steel, important physical properties of aluminum are the reduced modulus of elasticity, namely it is only about 30 % compared to that of steel, the reduced specific weight (also about 30 %), the reduced tensile strength (depending on the alloy), and the low melting point.

As to the other features, it is worthy to note the higher acquisition cost of the material and the difficulties with its processing (increased cost in man-hours due to special welding and further processing).

In addition, because of the low melting point, fire safety regulations prescribe a special thermal insulation for aluminum-alloy structures, which requires an overlay of aluminum walls, forming the borders of fire zones on board; this overlay is usually made of sheets of steel preventing the spread of fire to other zones.

Fig. 2.78 All aluminum alloy high-speed hybrid SWATH HSS1500 of STENA Lines

The connectivity/foundation of the aluminum-structure on the remaining steel structure (if any) requires special care, because of problems with welding (use of riveted joints with plastic insulation or use of contemporary cladding technologies).

Finally, it can be considered that with the use of aluminum alloys, for example, for deckhouses, the corresponding weight will be reduced by approximately 45–50%, while the resulting cost *per unit weight* can be 5–7 times higher than that of the corresponding steel construction (up to 10 times for shipyards with less expertise in aluminum processing) (Fig. 2.78).

F. Approximation formulas

1. Dry Cargo Ships

Wehkamp–Kerlen (Tech. Hochschule Aachen, 1985, for the weight of main hull, without superstructures)

$$W'_{ST} = 0.0832 \cdot A \cdot e^{-5.73 \cdot A \cdot 10^{-7}}$$
$$A = L^2_{PP} \cdot B \cdot C_B^{1/3}/12$$

Carreyette (Watson and Gilfillan 1976, RINA)

$$W_{ST} = C_B^{2/3}(LB/6)D^{0.72}\,0.002(L/D)^2 + 1$$

2. Tankers

Det Norske Veritas (1972)

$$W'_{ST} = \Delta[\alpha_L + \alpha_T(1.009 - 0.004\,L/B) \cdot 0.06 \cdot (28.7 - L/D)]$$

where

$$\alpha_L = \frac{(0.054 + 0.004 \cdot L/B) \cdot 0.97}{0.189 \cdot (100L/D)^{0.78}}$$

$$\alpha_{\mathrm{T}} = 0.0290 + 0.00235 \cdot \Delta \cdot 10^{-5}, \quad \text{for } \Delta < 6 \cdot 10^5 \text{ t}$$

$$\alpha_{\mathrm{T}} = 0.0252 \cdot (\Delta \cdot 10^{-5})^{0.3}, \quad \text{for } \Delta > 6 \cdot 10^5 \text{ t}$$

Limitations:

$$L/D = 10 \div 14$$
$$L/\mathrm{B} = 5 \div 7$$
$$L = 150 \div 480 \text{ m}$$

Assumptions:

- Use of mild steel
- Without superstructures/deckhouses
- Concerns old designs, without taking into account the influences of MARPOL, OPA90 and more recent CSR regulations.

Sato

$$W_{\mathrm{ST}} = \left[\frac{C_{\mathrm{B}}}{0.8}\right]^{1/3} \cdot \left[5.11L^{3.3}\frac{B}{D} + 2.56L^2(B+D)^2\right]10^{-5}$$

3. Bulk-Carriers

Det Norske Veritas (1972)

$$W_{\mathrm{ST}} = 4.274 \cdot Z^{0.62} \cdot L \cdot (1.215 - 0.035 \cdot L/\mathrm{B}) \cdot$$
$$\cdot (0.73 + 0.0025L/B) \cdot (1.0 + (L - 200)/1800) \cdot$$
$$\cdot (2.42 - 0.07L/D) \cdot (1.146 - 0.0163L/D)$$

where Z[m^3]: modulus of midship section

Limitations:

$$L/D = 10 \div 14$$
$$L/B = 5 \div 7$$
$$L = 150 \div 380 \text{ m}$$

Murray (Trans. IESS, 1965)

$$W_{\mathrm{ST}} = 0.0328 \cdot L^{1.65}(B + D + T/2) \cdot (0.5 \cdot C_{\mathrm{B}} + 0.4).$$

where L: length in foot (1 ft ≈ 0.3048 m)

4. Containerships

Chapman (Univ. of Newcastle upon Tyne, 1969)

$$W_{ST} = 0.0209 \cdot L_{PP}^{1.759} \cdot B^{0.712} \cdot D^{0.374}$$

Miller (Univ. of Michigan, 1968)

$$W_{ST} = 0.000435\left(L \cdot \mathrm{B} \cdot D\right)^{0.9} \cdot \left(0.675 + 0.5C_{B}\right) \cdot [0.00585\left(L / D - 8.3\right)^{1.8} + 0.939]$$

5. Various types of ships by Watson and Gilfillan

The following relationships were derived from the analysis of data of 70 (seventy) vessels of 14 (fourteen) different types.

$$W_{S} = W_{S1}(1 + 0.5\left(C_{B0.8D} - 0.70\right)$$

where

$$C_{B_{0.8D}} = C_B + (1 - C_B) \cdot \frac{0.80D - T}{3T}$$
$$W_{S1} = \mathrm{kE}^{1.36}$$
$$E = L\left(B + T\right) + 0.85L\left(D - T\right) + 0.85\Sigma(l_1 h_1) + 0.75\Sigma(l_2 h_2)$$

l_1, h_1 : length and height of superstructures
l_2, h_2 : length and height of deckhouses

Remarks

1. The basic form of all these formulas is:
   $W_{ST} = L^{a} \cdot B^{b} \cdot D^{c} \cdot C_{B}^{d} \cdot \mathrm{e}$.
   In some formulas, where C_B^{d} is missing, it is understood that the result is valid for characteristic block coefficients of relevant ship type.
2. All formulas are based on the metric unit system, unless otherwise indicated.
3. The accuracy of the formulas can be satisfactory (about ± 10 %), in all cases for which the ships under design do not differ significantly from the "standard" designs of the individual types. However, given that most of the above formulas were developed based on data of the 70s, the resulting weights can be relatively high for today's standards, in view of the general weight reduction due to the optimization of the structural weight with modern calculation methods and the extensive use of higher tensile steel (tankers, bulk-carriers).
4. All formulas can be easily programmed in computer codes for the optimization of the main dimensions in the preliminary design stage of a ship.

5. In all formulas with W'_{ST} denotes the weight of the steel structure of the main ship hull *without* the superstructures and deckhouses.

2.15.5 Weight of Equipment and Outfit

The weight of equipment and outfitting W_{OT} (Outfit Weight) of accommodation and overall ship arrangements, as defined in Sect. 2.15.1, generally includes the weight of all outfitting/equipment fitted to the "naked" ship hull, except for the machinery equipment.

In recent years we observe generally an increase of this weight category, mainly due to the improved quality of accommodation, for example, extension and enhancement of outfitting of crew's accommodation spaces, of sanitary facilities, of air-conditioning, and insulation against temperature changes and noise. The absolute increase of the weight of accommodation is not compensated by the incurred reduction of the crew number (for cargo ships).

As to the other equipment and outfitting beyond accommodation, a similar increasing trend is observed, particularly in comparison to data of the preceding 20 years, due to the increased weight of the cargo hold hatch covers (as applicable), the improved capabilities of cargo-handling means (higher lifting capacity of derricks and cranes), and the improved safety of firefighting facilities (CO_2-installations and insulations).

Certain structural components, such as stairways, derrick posts, rudder, steel hatch covers of holds, can be included either in W_{OT} or in W_{ST} following the practice of the yard or designer.

The incorporation of the various outfitting components to W_{OT} can be done in accordance to two general rules:

1. As to the *subject of work* of the various *production units* of the yard, for example machinery workshop, carpenter shop, etc. (see Table 2.30, for example).
2. As to the *functionality* of each element or group of elements (Table 2.31 of Schneekluth (1985), for instance).

The latter classification method facilitates the overall processing/production procedure in the shipyard, when ordering and installing the equipment: external suppliers/outsourcing, preparation of work/specification of equipment, construction/fabrication/acquisition/implementation-fitting/costing.

It is known that because of the nonuniformity/disparity of the W_{OT} elements it is not possible to develop unique methods for calculating the W_{OT}, as for the steel structural weight. In case of lack of comparative data from similar ships, one may resort to empirical formulas or coefficients for various types of vessels (see Tables 2.1 and 2.19), or diagrams from statistical data for specific types of ships.

Finally, the accurate calculation of the weights comprising the W_{OT} is only possible with the breakdown of the major outfitting weight groups, into individual weight components. The latter are estimated based on corresponding specifications of the shipyard

Table 2.30 Grouping of outfit weight components as products of corresponding shipyard's workshops or of external suppliers

I	*Heavy carpentry/wood work*: wooden decks, planking of holds, of refrigerated spaces and double bottom, wooden hatch covers, wooden bulkheads, wooden deckhouses, and nonwooden plating of holds (by aluminum or composite materials sheets)—contemporary specific weight values at the lower limit of Table 2.32
II_1	*Insulation work*: Insulation weight as a function of type of insulation material and less of insulation thickness. Typical values: $V_{\text{Net Net}}/LBD = 0.82–0.35$ or insulation weight/$V_{\text{Net Net}} = 30–80$ kp/m^3
II_2	*Coating and anticorrosion work*: coatings, paintings, asphalting, paving of floors, and walls
III	*Minor wood work*: internal accommodation walls, doors, furniture of accommodation spaces, carpeting of interior floors, curtains, upholstery, glass work. Typical specific weight/accommodation spaces' area: 60–70 kp/m^3
IV	*Piping works of ship*: piping for ballast, stripping, firefighting, freshwater-seawater, heating, scoopers, venting pipes, etc; all valves, bolts, etc; sanitary utensils, heating radiators; high values in the table for tankers and passenger ships due to extensive piping work
V	*Machining work*: steel doors, covers of hatches and bulkhead openings, etc.; stairs; machining work of interior accommodation arrangements, utensils for kitchen use and hotel functions (cookers, washing machines, etc.). Ducts for natural ventilation and air conditioning. Current values are at the upper limit of the table because of use steel hatch-covers; limited use of wood
VI	*Cargo handling equipment*: without masts (see steel structure), winches and derricks/cranes (see $VIII^2$), all the cargo handling components, namely derrick brackets, ropes, pulleys, hooks, chains, etc.; accurate estimation by specification of derrick/crane numbers, lifting capacity and external suppliers information
VII	*Towing and docking/mooring equipment*: except for the winches (see $VIII^2$), all towing and docking/mooring equipment. The given values in the table decrease with the absolute size of the ship
$VIII_1$	*Refrigeration equipment*: for reefer cargo spaces
$VIII_2$	*Other auxiliary machinery*: rudder gear, winches for all uses (anchors, loaders, life-boats), air conditioning, firefighting. Electrical installations. Communication facilities. High values in the table for cargo ships with heavy lifting equipment, refrigerated spaces; also, high values for passenger ships due to the extensive installations of electrical, air conditioning, firefighting, and communication equipment
	Only for electrical installations: cargo ships: 0.8–1.4 kp/m^3, tankers: 0.7–1.0 kp/m^3, reefer ships: 1.0–1.5 kp/m^3, passenger ships: 3–4 kp/m^3; out of these weight values, 50–80 % concern the weight of cables
	Weight of refrigeration units for cargo spaces depends on the net volume to be cooled: Weight/$V_{\text{Net Net}} = 20–30$ kp/m^3
IX	*Other equipment*: anchors, chains, ropes, canvas, life-boats, navigation marking equipment, tools, supplies, kitchenware, mobile equipment for accommodation spaces—high values for passenger ships

or relevant information of external suppliers (detailed design phase). Certainly, this work is very laborious and usually the final outcome does not reach the accuracy of the steel or machinery weight estimations. However, the implementation of modern computerized systems in the production process of shipyards enables the recording, classification, and post-processing of individual outfitting items relatively easily (Table 2.29).

Table 2.31 Grouping of tasks and components of the ship's construction and outfitting in accordance with the function/operation of each component by Schneekluth (1985 in German)

0 OBJEKTKOSTEN ALLGEMEIN	1 SCHIFFSKÖRPER	2 AUSRÜSTUNG ZUR FAHRBEREITSCHAFT	3 AUSRÜSTUNG Z. NUTZUNG	4 EINRICHTUNG	5 VORTRIEB	6 VERSORGUNG WASSER+LUFT
01 Konstruktive Arbeiten 012 Entwurf 013 Modellversuche 014 Entwickl.-Kosten	11 Einzelteile Untergegliedert nach Bauelementen	21 Steuern 211 Ruder 212 Rudermaschine 218 Rohrleitungen + Kleinteile	31 Trockenladung 311 Wegerung 313 Sondereinr. für Container 314 für Decksladung 318 Rohrl.+Kleint.	41 Komplette Räume 411 für Betrieb 412 Wohnen Besatz. 413 Wohnen Passag. 416 Sanitär 417 Wirtschaftsr. 419 Treppenhäuser	51 Dieselanlage 511 Hauptmotor 512 Luft+Abgas 513 Kraft+Schmierst. 515 Kühlung 516 Fahrstand 518 Rohrl.+Kleint.	61 Seewasser 611 Pumpen 612 Drucktanks 613 Filter 614 Peilen 618 Rohrl.+Kleint.
02 Fertig.-Vorbereitung 021 Schnürboden 022 Modelle	12 Zubehörteile - örtl. Verstärkungen - Begrenzungen - Konsolen - lose Tanks - Kleinfundamente	22 Festmachen 221 Ankereinrichtung 224 Verholeinrichtung 228 Rohrleitungen + Kleinteile	32 Kühlladung 321 Kühlanlage 322 Isolierung 323 Raumausstattung 328 Rohrl.+Kleint.	42 Trennwände 421 Verschalungen 424 Trennwände 425 Türen 427 Isolierungen 428 Kleinteile	52 Dampfanlage	62 Frischwasser 621 Pumpen 622 Drucktanks 623 Filter 624 Peilen 428 Rohrl.+Kleint.
03 Bauplatz-Kosten 031 Helling 032 Kai		23 Rettung 231 Boote 232 Aussetzen 238 Rohrleitungen + Kleinteile	33 Ladegeschirr 331 Masten + Bäume 333 Takelung 334 Winden 335 Kräne 338 Rohrl.+Kleint.	43 Möbel 431 Schränke 433 Tische 434 Sitzmöbel 437 Kojen 438 Kleinteile		63 Abflüsse 631 Pumpen 633 Siebe,Filter 634 Peilen 638 Rohrl.+Kleint.
04 Hilfsarbeiten 041 Transporte 042 Heben 043 Reinigung 044 Bewachung		24 Luken 241 Lukendeckel 244 Mannlochdeckel 246 Klappen 247 Oberlichter 248 Kleinteile	34 Oelladung 341 Oelpumpen 342 Wasserpumpen 343 Dampf 344 Lüftung 347 Tankreinigung 348 Rohrl.+Kleint.	44 Sanitäreinr. 441 WC/Urinal 443 Badewannen 444 Duschen 446 Waschbecken 448 Kleinteile		64 natürl.Lüftung
05 Stapellauf 051 Berechnung 052 Durchführung		25 Verkehr 251 Leitern+Treppen 253 Geländer 255 Grätinge 256 Sonnensegel 258 Kleinteile	35 Sonderladung 351 Gas 352 Zement 353 Chemikalien 358 Rohrl.+Kleint.	45 Verpflegung 451 Kochen,Braten 454 Kühlschränke 456 Proviant-Kühlr. 457 Provianträume 458 Rohrl.+Kleint.		65 künstl.Lüftung 651 Laderäume 652 Sonstige Schiffsbereiche
06 Kontrolle 063 Bord-Erprobung 064 Bauaufsicht		26 Fenster + Türen 261 Fenster 262 Bullaugen 264 Türen 267 Schottüren 268 Kleinteile	36 Sonderaufgaben 361 Bergen 362 Schleppen 368 Rohrl.+Kleint.	46 Hilfsgeräte 461 Geschirrspülen 462 Abfall 464 Wäscherei 467 Gepäck 468 Rohrl.+Kleint.	56 Leistungsübertr. 561 Welle 562 Getriebe 564 Kupplung 565 Kühlung 568 Rohrl.+Kleint.	66 Klimaanlage 668 Rohrleitungen + Kleinteile
07 Ablieferung 071 Probefahrten 072 Dokumente 075 Überführung 076 Einweisungspersonal		27 Feuerlösch 271 Wasser 272 Dampf 273 CO_2 274 Schaum 278 Rohrleitungen + Kleinteile		47 Betreuung 471 Messen 472 Speisesäle 473 Schwimmb./Sport 475 Läden 476 Medizin.Versorg. 478 Rohrl.+Kleint.	57 Vortrieb 571 Festpropeller 572 Verstellprop. 573 Sonderprop. 578 Rohrl.+Kleint. 579 Verschiedenes	67 Heizung 678 Rohrleitungen + Kleinteile
08 Verwaltung 081 Finanzierung 082 Provisionen Honorare 083 Versicherung 084 Patente		28 Materialschutz 282 Konservierung 283 Verzinken 284 Farbanstrich 288 für Rohrleitungen 289 Decksbelag außen		48 Isolierungen 481 Feuer,Wärme 482 Schall 483 Masch.-Raum 484 Decksbeläge innen 488 Kleinteile		

Table 2.31 (continued)

7 ENERGIE-ERZEUGUNG	8 SCHIFFSFÜHRUNG	9 INVENTAR
71 Elektrisch	81 Lichter,Signalanlagen	91 Zimmermann
711 Generatoren 713 Gleichrichter 715 Stromspeicher 718 Kleinteile	811 Lichter 814 Signale 819 Kleinteile	911 Leinen 912 Allg. Inventar 913 Verbrauchsstoffe 916 Materialmitgaben 917 Werkzeuge 919 Unterbringung
72 Hydraul.+Pneum.	82 Navigation	92 Rettung
721 Hydraulik 723 Pneumatik 728 Rohrleitungen	821 Kompasse 822 Selbststeuer-Anlagen 823 Funknavigation 828 Rohrleitungen + Kleinteile	921 Rettungsinventar 922 Feuerlöschinventar 926 Werkzeug 929 Unterbringung
73 Dieselantrieb	83 Kommando-Anlagen	93 Sonderinventar
731 Motoren 732 Luft + Abgas 733 Kraftstoff 735 Kühlung 738 Rohrl.+Kleint.	831 Sprechanlage 833 Alarmanlage 838 Kleinteile	
74 Dampfantrieb	84 Funkanlage	94 Wirtschafts-Inventar
	841 Funkanlage 842 Fernsehen 843 Telefon	941 Deck 942 Messen 943 Kammern 944 Reinigung 947 Werkzeuge 949 Unterbringung
75 Schaltanlage	85 Rufanlage	95 Maschine
751 Schalttafeln 752 Verteiler 753 Schaltgeräte 754 Meßgeräte 758 Kleinteile	851 Kammer-Rufanlage	951 Instrumente 953 Verbrauchsstoffe 954 Werkzeug + Gerät 955 Ersatzteile 959 Unterbringung
76 Kabelnetz	86 Überwachung	
761 Stromerzeugung 762 Stromverteilung n. Abschnitten 768 Kleinteile	861 Masch.-Überwachung 862 Fernmeß-Anlagen für Tanks + Bunker 863 Sonstige Überwachung 868 Rohrl.+Kleinteile	
77 Beleuchtung	87 Automation	97 Elektriker
771 Hauptbeleuchtung 772 Notbeleuchung 778 Kleinteile	871 Überwachung 872 Fernbedienung 878 Rohrleitungen + Kleinteile	971 Instrumente 976 Werkzeuge 979 Unterbringung
78 Abgas-u.Hilfskess.		98 Nautik
		981 Nautisches Inventar 982 Laternen 983 FT-Inventar 989 Unterbringung

Explanations: *0* general cost items (studies, preparation of production process, launching, ship delivery, and administration), *1* outline of ship hull components, *2* outline of outfitting for ship operation, *3* outline of outfitting for servicing the payload, *4* accomodation, *5* propulsion, *6* supply of water and air, *7* power generation, *8* steering and navigation, *9* spare parts, tools, utensils for accomodation, etc.

Table 2.32 Specific weight coefficients w for outfitting components w=weight/L B D [kp/m^3], D: side depth of strength deck (see Table 2.30) for ordinary merchant ships by E. Strohbusch (1971)

Ship type	Cargo	Tanker	Reefer	Passenger
Group				
I	1.5–6	0.5–1	1.5–5	8–14
II$_1$	–	–	10–26	–
II$_2$	4–7	1–2	4–7	4–10
III	5–6	1–2	6–8	8–12
IV	1.2–1.5	2.5–5	1.2–1.5	5–6
V	2–4	1.5–2	2–4	10
VI	2.5–4	0–0.1	1	0.5
VII	1–1.5	0.3–0.5	1–1.5	1
VIII$_1$	–	–	6.5–10	–
VIII$_2$	4–7	1.5–2	4–7	12–20
IX	2–3	1–1.5	2–3	3–4

A. **Use of coefficients**

In case of lack of other data from similar ships, the designer may use empirical coefficients, as in the listed tables (Tables 2.1, 2.32, and Papanikolaou and Anastassopoulos 2002), or the references mentioned below.

These coefficients depend mainly on the ship type, on ship size and the outfitting quality. Of course, the employed coefficients should be appropriately adapted to the characteristics of the ship in such a way that they remain nearly constant for ordinary sizes of each ship type.

Provided that there are approximate data from similar ships available, their adaptation to the subject ship can be done by use of relational coefficients, as outlined in Appendix C (relational method of Normand).

Though outdated, the main references in the open literature regarding the appropriate use of coefficients for the calculation of W_{OT} are the following:

a. *Henschke*, Vol. 2, p. 465: Adapted coefficients to be multiplied by ($L \cdot B \cdot D$).
b. *Weberling*, Handbuch der Werften (HdW), Vol. VII, p. 50–52 and HdW, Vol. V III, p. 144 (tankers and reefers)
c. *Watson-Gilfillan*, RINA 1976: Adapted coefficients to be multiplied by $L \cdot B$ instead of $L \cdot B \cdot D$
d. *Krause, in Henschke*, Vol. 2, p. 94: Adapted coefficients referring to the holds volume ∇_c; reference to the analysis of the main groups of W_{OT}
e. *Danckwardt*: Adapted coefficients referring to the holds volume $_c$, the deadweight DWT and the number of crew (see Figs. 2.79, 2.80, and 2.81).
g. *Henschke*: Adapted coefficients to be multiplied by $(L \cdot B \cdot D_{SS})^{2/3}$

where D_{SS} means the corrected side depth D, which accounts for the average height of the superstructure. The latter corresponds to the superstructure volume divided by the deck area.

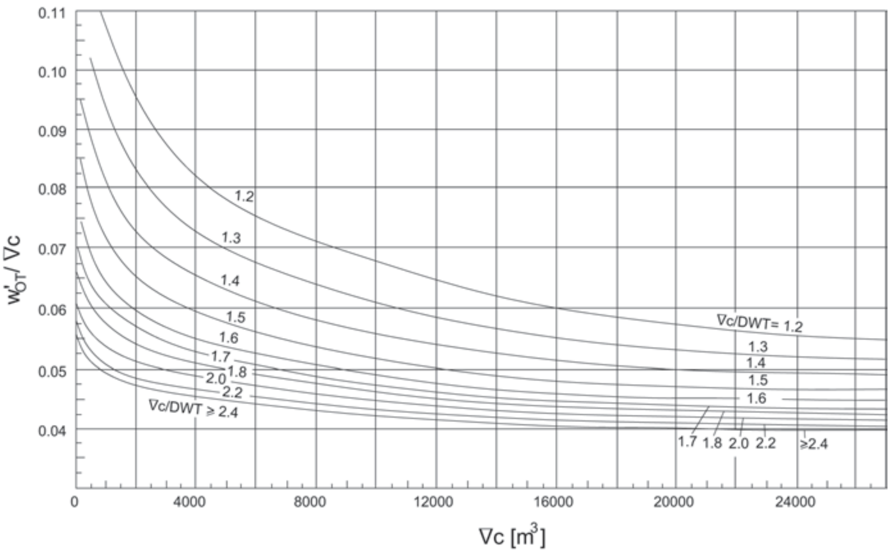


Fig. 2.79 Weight of outfitting versus the hold volume c and the ratio c/DWT for dry cargo ships according to Henschke (1964)

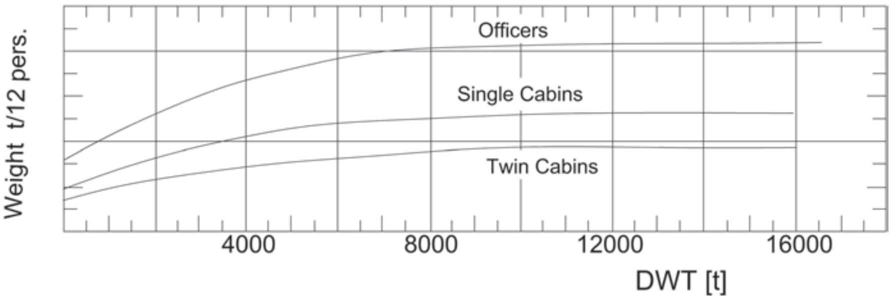


Fig. 2.80 Weight of accommodation outfit dependent on crew seniority vs. DWT for cargo ships by Henschke (1964)

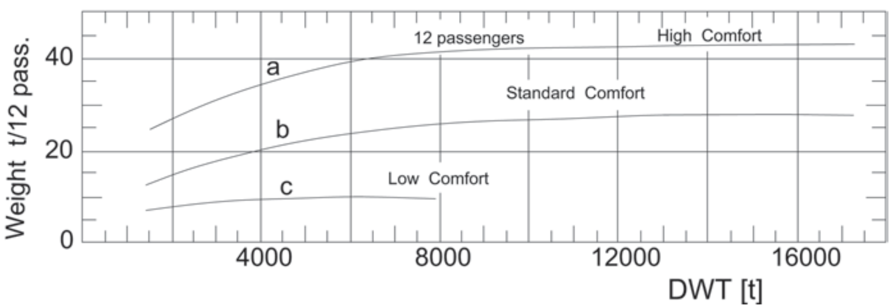


Fig. 2.81 Additional weight of accommodation outfit for 12 passengers vs. the DWT by Henschke (1964)

B. **Approximate formulas (preliminary design stage)**
Cargo Ships

$$W_{OT} = K_{OT} \cdot L \cdot B$$

where

$$\begin{aligned} K_{OT} &= 0.40 - 0.45 \text{t/m}^2 \text{(general cargo ships)} \\ &= 0.34 - 0.38 \text{t/m}^2 \text{(container ships)} \\ &= 0.22 - 0.25 \text{t/m}^2 \text{(bulk-carrier, } L \cong 140\text{m)} \\ &= 0.17 - 0.18 \text{t/m}^2 \text{(bulk-carrier, } L \cong 250\text{m)} \\ &= 0.28 \text{t/m}^2 \text{(tanker, } L \cong 150\text{m)} \\ &= 0.17 \text{t/m}^2 \text{(tanker, } L \cong 300\text{m)} \end{aligned}$$

Dry cargo ships according to Henschke-Schneekluth (see Fig. 2.50, without accommodation)

$$W_{OT} = \frac{0.07(2.4 - \nabla_c / \text{DWT})^3 + 0.15}{1 - \log_{10} \nabla_c} \cdot \nabla_c$$

where

∇[m³]: hold volume (Grain)
∇/DWT [m³/t]: capacity factor.

This formula is valid for capacity factors in the range of:

$$1.2 \leq_c \nabla / \text{DWT}[\text{m}^3/\text{t}] \leq 2.4.$$

Reefer cargo ships according to Carreyette (Transaction of Royal Institute of Naval Architects 1976, p. 134)

$$W_{OT} = A \cdot (L/100)^2 + B(\nabla_i / 1000)^{2/3}$$

where

L: length between perpendiculars
∇_i: total gross volume of reefer spaces/holds
A = 550
B = 163

Assumptions (Reefers)

- L = 90 ÷ 165 m
- Ships built in the 60s

Passenger ships (without vehicles, passengers in cabins)

$$W_{\mathrm{OT}} = K_{\mathrm{OT}} \cdot \Sigma_i \nabla i$$

where

$K_{\mathrm{OT}} = 0.036 \div 0.039 \mathrm{t/m}^2$
$\sum_i \nabla \mathrm{i} =$ total gross registered volume (GRT)in[m^3].

RoPax-Passenger Ships

The above coefficient K_{OT} is modified for passenger/RoPax ships and passenger ships of restricted voyages (without cabins) as follows:

$$K_{\mathrm{OT}} = 0.04 \div 0.05 \mathrm{t} / \mathrm{m}^3$$

C. **Use of approximate diagrams**

The outfit weight W_{OT} of cargo ships can be also approximated by analyzing it into one part which is dependent on the size of the ship, for instance, the hold volume or the DWT, and another one that refers to the number of crew or the specific requirements of the owner.

For dry cargo ships, the first weight part of W_{OT} can be obtained from Fig. 2.79 as a function of the hold volume ∇_{C} and the ratio ∇_{C}/ DWT (see Henschke 1964).

Herein, we assume an ordinary ship with two decks, steel hatch covers on the uppermost deck and wooden cover for the intermediate deck. Correction for a third deck will be: +5–10 %. Likewise, corrective increases are required for ships with extra wide hatch covers, which also require larger, non-wooden covers for the intermediate deck openings.

The second part of W_{OT} that depends on the number of persons on board (crew and possible passengers) can be obtained from Figs. 2.80 and 2.81 that account for the quality of accommodation for the persons on board.

The below Fig. 2.82 provides the ratio of W_{OT} to $L_{\mathrm{BP}} \cdot B$ as a function of length L_{BP} for various types of ships, while from Fig. 2.83 the W_{OT} can be obtained as a function of the product $L \cdot B$ for passenger ships.

Similar diagrams also exist for other types of ships, such as tankers and bulk-carriers (see e.g., Lewis 1988; Henschke 1964), however, the more outdated data in Henschke (1964) are inferior to those resulting from application of the foregoing methods (A and B) in terms of accuracy.

D. **Detailed calculation of groups of outfit weights**

The following W_{OT} estimation method was proposed by Schneekluth (1985); it forms an intermediate approach in between the detailed calculation of the individual outfit weights and the approximate methods (A to C). The accuracy of the method

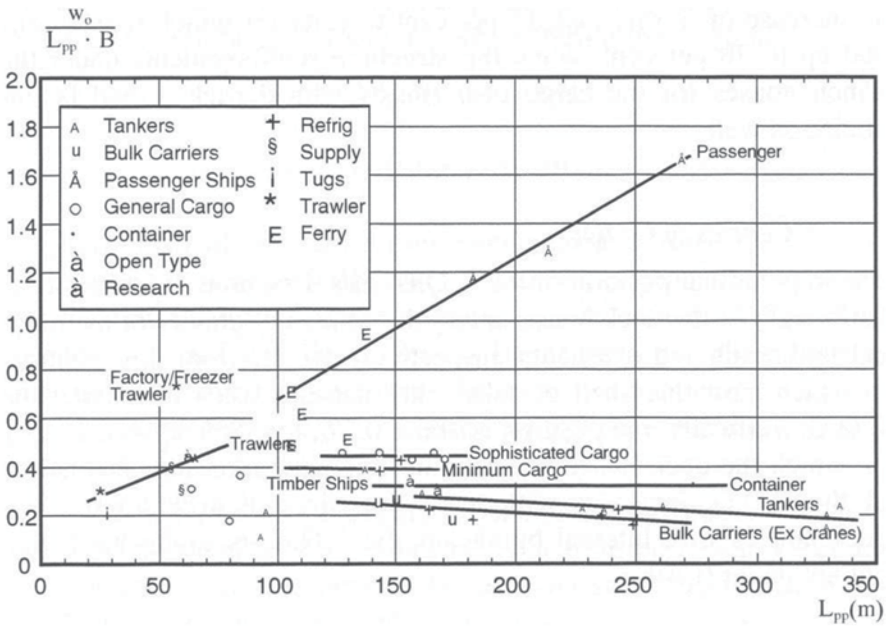


Fig. 2.82 Ratio of outfit weight to $L \cdot B$ as a function of length L by Watson (1998). (in Friis et al. 2002)

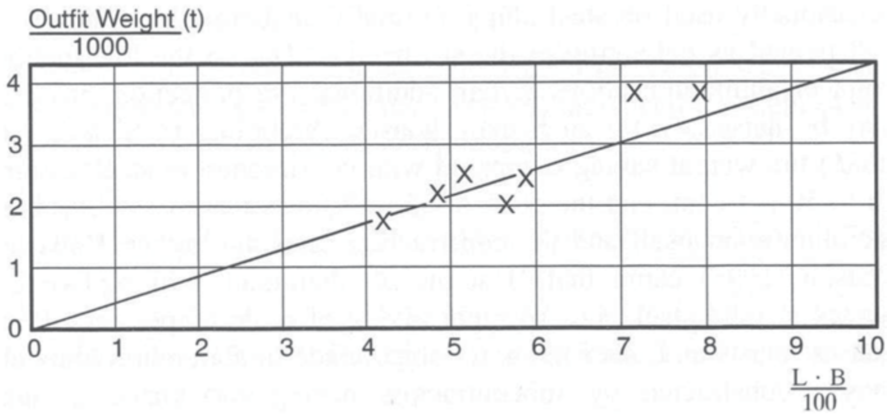


Fig. 2.83 Outfit weight as a function of $L \cdot B$ for passenger ships by Watson (1998). (in Friis et al. 2002)

is satisfactory for all design stages, beyond the preliminary phase. Besides the calculation of weights, this method also facilitates the estimation of the weight centers.

The main principles of the method are:

1. Certain groups of weights of W_{OT}, distinguished by their relatively large absolute weight (e.g., hatch covers, loaders, etc.), can be calculated accurately from the very beginning, avoiding approximation errors by use of empirical coefficients.
2. Coefficients are used only for those groups of weights of W_{OT}, for which the conceptual reduction to certain characteristic sizes of the ship, for example, the accommodation area, is possible and known, without large uncertainty. In addition they can be used for onboard equipment that is independent of ship type.
3. If several weight subgroups are calculated approximately, there is a high probability that the errors in the individual estimations are heterogeneous as to their sign. Thus, compared to an approach referring the total W_{OT} through coefficients, one may expect a balancing of differences resulting from the individual estimations (errors of opposing signs partially cancelling each other). The method applies primarily only to general cargo ships and containerships; however, the extension to other types of ships with corresponding adaptation of required changes appears possible.

D1. *Weight groups of W_{OT} by Schneekluth*

I. Hatch covers
II. Cargo-handling equipment
III. Accommodation
IV. Other weights.

D2. *Approximations of weight groups*

I. Hatch Covers: This group includes *all* weights of the hatch covers, and their built-in driving system (Table 2.33; Figs. 2.84 and 2.85).

Malzahn's Formula for the Single–Pull system with a load of 1.75 t/m^3

Table 2.33 Weight of weathertight Single-Pull hatch covers versus hatchway size and vertical loading due to deck-containers.

	Weight [kp] per meter of hatchway length				
Hatchway breadth [m]	6	8	10	12	14
Normal load 1.75 t/m[a]	826	1,230	1,720	2,360	3,150
Load by one layer of containers[a]	826	1,230	1,720	2,360	3,150
Load by two layers of containers	945	1,440	2,010	2,700	3,550

[a] 20 ft (TEU) containers are assumed having a weight of 20 t.
[b] For the "Piggy Back" system reduction of weights by about 4 %

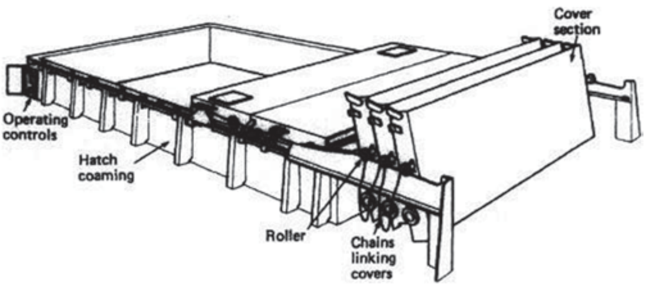
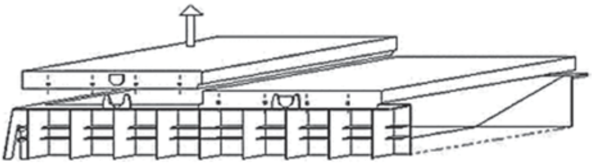


Fig. 2.84 Single-pull weather-deck hatch cover

Fig.2.85 Piggy-back hatch cover



$$W_H / l_H = 0.0533 \cdot b_H^{1.53} + \delta b_H \cdot 0.065$$

where

W_H: weight of cover [t]
l_H: length of cover [m]
b_H: breadth of cover [m]
δb_H: difference in breadth beyond 12 m.

For pontoon type covers, the weight estimated by the formula of Malzahn can be reduced by up to approximately 15 % (Table 2.34; Fig. 2.86).

Table 2.34 Weight of nonweathertight hatch cover of foding type

	Weight [kp] per meter of hatch breadth				
Breadth of hatch [m]	6	8	10	12	14
Normal load[a]	845	1,290	1,800	2,440	3,200
Use of forklift[b]	900	1,350	1,870	2,540	3,360
Two layers container[c]	930	1,390	1,940	2,600	3,460

[a] Normal load for a deck height up to 3.5 m (GL).
[b] Forklift trucks of a total weight of 5t, with rubber wheels.
[c] 20 ft container (TEU) and 20 t/TEU.
[d] The total weight of the hatchway covers on general and multi-purpose cargo ships or semi-containerships can reach values of up to 50% of W_{OT}.

Fig. 2.86 Folding type tween deck hatch covers

Tween decks (Nonweathertight covers—folding type design**)**

II. Cargo-handling equipment: This group includes: Derricks, winches, deck cranes, planking of hold, lashing units of containers; however, *without* the derrick's mast that is typically included in W_{ST}. For Ro-Ro ships, all the ramps, external or internal, are included in this subgroup of weights.

Lightweight of derricks and cranes[25] (Fabarius, Handbuch der Werften, Vol. VII, p. 168 Henschke, Vol. 2, p. 97)

Weights are functions of lifting capacity and boom length. For rotating cranes, the following applies(Table 2.35; Figs. 2.87 and 2.88):

Table 2.35 Weight of rotating cranes by Fabarius. (Schneekluth 1985)

Maximum lift weight [t]	Maximum span [m]	Structure's height [m]	Crane Weight [t]
1	10	3.7	10
2	10	3.7–4.3	7–11
–	14	4.3–5.0	8–13
3	10	3.7–4.5	8–11
–	16	4.3–5.0	10–15
5	10	3.7–5.1	10–15
–	16	4.7–6.3	13–16
7.5	14.5	5.9	20
–	16	6.5	21

[25] A *derrick* is a lifting machine for hoisting and moving heavy objects, consisting of one or more movable booms equipped with cables and pulleys and connected to the base of an upright stationary mast. The movements of the boom (up-down-sideways-lift of weight) are supported by winches. A *crane* is a contemporary development of the derrick; in difference to the derrick, the movement of the boom is enabled by its turning base and the hoisting and moving of objects by means of cables attached to the boom.

Fig. 2.87 Outdated and contemporary general cargo ships equipped with conventional derricks (*left*) and turning cranes (*right*) respectively

Fig. 2.88 Heavy lift Stülcken derrick®

Heavy lift derricks

The weights of derricks and cranes are generally functions of their lifting capacity, lifting speed and type of winches. Approximate values: 0.16–1 t per t of lifting capacity. More detailed descriptions and data may be found in Papanikolaou and Anastassopoulos (2002).

Planking of holds

Modern cargo ships are constructed without interior planking of the holds unless required by the owner. However, for the planking of the sides of hold spaces, with wooden planks, the required wood volume can be approximated by the projected area of the hold multiplied by a mean thickness of 50 mm. The same can be applied to the planking of the bulkheads. In the calculated weight a margin of 10 % is added for the fittings.

For the planking of hold's floor, usually pinewood is used, namely longitudinal planks of thickness 80 mm can be fitted, which are supported at each frame by transverse battens, of 40 mm × 80 mm cross section.

Lashing units of containers

For containers on deck the weight of lashing equipment needs to be added, that is (Figs. 2.89, 2.90, and 2.91),

$$\begin{aligned} &0.024\ \text{t/TEU}\ \left(\text{container } 20'\right) \\ &0.031\ \text{t/FEU}\ \left(\text{container } 40'\right) \\ &0.043\ \text{t/TEU}\ \left(\text{mixed loading with TEU and FEU}\right) \end{aligned}$$

Ramps of Ro-Ro ships

Exterior ramps

Fig. 2.89 Container lashing

Fig. 2.90 Ro-Ro loading ramp

Fig. 2.91 Ro-Ro interior ramp

$$\text{length } 5\text{ m: } \sim 0.3 \div 0.4\ \text{t/m}^2$$
$$20\text{ m: } \sim 0.4 \div 0.6\ \text{t/m}^2$$
$$50\text{ m: } \sim 0.55 \div 0.75\ \text{t/m}^2$$

Interior ramps

$$\text{length } 15\text{ m: } \sim 0.15 \div 0.25\ \text{t/m}^2$$
$$50\text{ m: } \sim 0.30 \div 0.40\ \text{t/m}^2$$

III. Accommodation: This group of weights referring to the accommodation quarters of crew and passengers, includes:

- Separation walls of superstructures, if not included in W_{ST}

- Panelling/insulation of interior rooms
- Sanitary installations and related pipes
- Doors, windows, other coverings of openings
- Heating, ventilation, air conditioning
- Kitchenware and other household utensils
- Furniture and arrangements of spaces
- Lighting and cables

All the weights included in this group can be calculated through the corresponding volume of the fitting or through the respective accommodation area. Characteristic values are:

Small to medium size cargo ships

$$160 \text{ to} 170 \text{ kp/m}^3$$
$$\text{or} \quad 60 \text{ to} 70 \text{ kp/m}^2$$

Large cargo ships, tankers

$$180 \text{ to} 170 \text{ kp/m}^3$$
$$\text{or} \quad 60 \text{ to} 70 \text{ kp/m}^2$$

Notes

1. The above specific weights generally increase for improved accommodation quality.
2. The values also increase for ships of absolutely large size and for corresponding very large accommodation areas (e.g., mega cruise ships).
3. For passenger ships, the values depend directly on the quality of the passengers' accommodation; the use of data from similar ships is essential.

IV. Other weights
The following items belong to this group:

- Anchors, chains, hawsers
- Anchor-handling and mooring winches, bollards
- Steering mechanism (excluding rudder)
- Refrigeration equipment
- Insulating works beyond interior accommodation
- Rescue equipment and launching systems
- Bulwarks, stairs, doors and covers beyond indoor accommodation area
- Fire-fighting systems
- Pipes, bolts, valves, gauges (outside the engine room and accommodation area)
- Hold ventilation
- Navigation facilities and signalling equipment

- Tools of deck crew

As in the previous category, the weight of this group is mainly a function of the ship size; it is independent of ship type.

Approximation formulae

$$W_{\mathrm{IV}} = (L \cdot B \cdot D)^{2/3} \cdot C_1, \text{where } C_1 = 0.18 \div 0.26 \text{ or}$$
$$W_{\mathrm{IV}} = W_{\mathrm{ST}}^{2/3} \cdot C_2, \qquad C_2 = 1.0 \div 1.2$$

where, W_{ST} και W_{IV} are given in [t] and L, B, D in [m]

General comments

The present method of splitting the W_{OT} into four subgroups can be modified for other ships than general cargo types of ships, such as reefers and tankers, by creating additional subgroups for the reefer cargo holds and the piping system of tankers, respectively.

E. **Centre of weights of W_{OT}** (Weberling, Handbuch der Werften, Vol. VII, p. 56 & Vol. VIII, p. 138)

General principles

1. If the weight components of outfitting were calculated individually, for example, by method type D or even in a more elaborate way, then the mass centre of the group W_{OT} can be estimated through the balance of the sum of the individual moments.
2. If the weight W_{OT} has been approximated globally, then it can be further analysed by breaking it down into subgroups and by taking the corresponding moments following method A.
3. If there are data from similar ships for the W_{OT} group, they can be used as first approximations.
4. Typical values for the vertical mass centre of the W_{OT} group

   Dry cargo ships:
   $KG_{\mathrm{OT}} = (1.00 \div 1.05) \cdot D_{\mathrm{SS}}$
   Tankers:
   $KG_{\mathrm{OT}} = (1.02 \div 1.08) \cdot D_{\mathrm{SS}}$

   where the corrected side depth D_{SS} was already defined before.

5. For the initial estimations, relevant tables of reference Papanikolaou and Anastassopoulos 2002 (see also Table 2.19) are very useful.

2.15.6 Weight of Machinery Installation

The weight of the machinery installation, which can be decomposed (see definition, Sect. 2.15.1) into:

Table 2.36 Weight W_{MM} for various types of main engines of merchant ships. The power given in the table is the maximum continuous rating (MCR)

Type of engine	Power (kW)	Weight (t/kW)	RPM
Slow-speed diesel	2,000–5,000	0.015–0.022	250–175
	5,000–10,000	0.022–0.029	175–100
	10,000–70,000 (84,420[a])	0.029–0.039	100–80
Medium-speed diesel	600–17,000 (20,000)	0.009–0.018	900–400
High-speed diesel (MTU type)	240–9,100	0.003–0.004	$>1{,}000$
Gas turbines (LM type)	4,412–42,160	0.001	$>3{,}600$

[a] The world's largest diesel engine in the year 2010 was the Wärtsilae-Sulzer RTA96-C marine diesel engine of about 84,420 kW (113,210 HP) @ 102 RPM delivered horsepower

$$W_M = W_{MM} + W_{MS} + W_{MR}$$

where

W_{MM}: weight of main engine
W_{MS}: weight of shaft and propeller
W_{MR}: weight of rest mechanical components,

includes the following weights:

- Main engine installation, consisting of the main engine(s) with reduction gear units (*only for non-low-speed diesel engines*), or of turbines with boilers (W_{MM})
- The exhaust system (W_{MR})
- The propellers and the transmission system, that is, propeller shaft(s) and shaft bearings, including stern–tube bearing (W_{MS})
- The electric generators, the cables to the switchboards/transformers (W_{MR})
- Pumps, compressors, separators (W_{MR})
- Pipes in the engine room (with fillings), also (often) piping of double bottom for pumping fuel or ballast (W_{MR})
- Desalination/drinking water production equipment (W_{MR})
- Sewage disposal system (W_{MR})
- Other equipment of the engine room: ladders, floor gratings, heat and noise insulations (W_{MR})
- In addition, usually: central refrigeration facilities (for reefer ships); outfitting of cargo pump room (tankers; W_{MR}, if not included in the W_{OT})

Factors affecting the weight of machinery installation

1. *Type of main engine*: Diesel of slow-speed, medium-speed, high-speed (small vessels), diesel-electric propulsion, steam turbine, gas turbine (mainly for naval ships); affects W_{MM}; (Table 2.36)
2. *Ship type and type of carried cargo,* for example, passenger ships and reefer cargo ships have a high demand on electrical energy (high W_{MR}). Also, diesel engine-powered tankers need a special boiler to produce steam for the cargo discharging pumps, the heating of cargo and cleaning of tanks (affects W_{MR}).
3. *Number of propellers* (affects W_{MS})

4. *Position of engine room* (affects W_{MS}, because of the length of the propeller shaft)
5. *Owner's special requirements* concerning the disposition of backup machines/ components, electric generator sets, etc

Methods of calculating weight W_M

A. Approximation of the total weight of W_M or the subgroups W_{MM}, W_{MS}, W_{MR} based on empirical coefficients (initial study)
B. Calculation based on known individual weights that constitute the W_M (final design phase)
C. Calculation based on comparable data of similar engine installations (initial study)
D. Approximation leading to a relationship to the weight of the main engine (initial study)
E. Calculation based on a breakdown of W_M into subgroups (advanced stage of design study)

A. **Approximation method based on empirical coefficients (initial study)**

During the preliminary design stage, W_M can be approximated through empirical coefficients referring to the W_{MM}, W_{MS}, and W_{MR} subcomponents that make up the W_M (see Table 2.37). These coefficients, which refer to the various types of ships, are normalized partly by use of the installed propulsion power (W_{MM} and W_{MS}) or by the volumetric product $L{\cdot}B{\cdot}D$—alternatively the propulsion power—for the W_{MR} weight.

B. **Calculation Based on Known Individual Weights**

In the final design stage, rarely for merchant ships, but extensively in the study of naval ships and submarines, the weight of the engine installation is calculated by summing up all individual weights that make up the W_M. During this laborious work the following points must be taken into account:

1. In the weight of pipes, boilers, and settling tanks located in the engine room, which comprise (W_{MR}), the weight of contained liquids (water, oil, and lubricants) must be added.
2. Particularly, as to the weight of the rest machinery installation (W_{MR}), all individual weight components of the engine room equipment must be added.

C. **Calculation Based on Comparable Data of Similar Machinery Installations**

Provided that comparable data of similar engine plants are available, we must pay attention to the following points:

- Type of main engine (diesel, turbine, etc.)
- Subtype of main engine (diesel engine cylinders "in serial arrangement" or V-type, steam pressure turbine)
- Number of revolutions of engine and propeller

Table 2.37 Coefficients of weight groups of machinery installation for merchant ships according to E. Strohbusch (1971)

Ship type	Cargo ship	Tanker	Reefer ship	Fast passenger ocean liner	Fast small passenger ship
Coefficient					
w_1 (kp/m^3)	10–15	3–5	20–25	15–25	25–45
w_2 (kp/HP)	35–50	25–35	50–70	20–30	30–55
w_3 (kp/HP)	5–10	4	8–10	8	5–10
w_4 (kp/HP)	Low-speed diesel engine 30–40	Steam turbine 20–25	Low-speed diesel engine 30–40	Steam turbine 20–25	Medium-speed engine with gearbox: 22–30 New technology: 12–17
w_5 (kp/HP)	85–90	55–60	90–110	50–60	70–80

1. Analysis of machinery weight:

$$W_{\mathrm{M}} = W_{\mathrm{MM}} + W_{\mathrm{MS}} + W_{\mathrm{MR}}$$

W_{MM}: weight of main engine(s) and gearbox(s) (for turbo machinery: turbines, gearbox, boilers)
W_{MS}: weight of propeller shaft and propeller(s) (includes: all shaft bearings, including crankshaft and stern-tube bearings)
W_{MR}: weight of rest machinery installation components (support equipment for the operation of main engine: fuel pumps, pumps for lubrication oil, cooling water, evaporators, etc. Piping of engine room for fuels, lubricants, cooling, steam, etc. Exhaust ducts, funnel. Boilers. Ventilation ducts of engine room. Mobile tanks of engine room, pumps for ballast, stripping, firefighting, engine room fresh water. Main electrical installation, electric generators, transformers, switchboards. Engine room tools.

2. Definitions: $w_1 = W_{\mathrm{MR}}/LBD$, $w_2 = W_{\mathrm{MR}}/\mathrm{SHP}$ (SHP: shaft horse power), $w_3 = W_{\mathrm{MS}}/\mathrm{SHP}$, $w_4 = W_{\mathrm{MM}}/\mathrm{SHP}$, $w_5 = W_{\mathrm{M}}/\mathrm{SHP}$

- Size of ship and engine room
- Magnitude of propulsion power
- Magnitude of electrical power generation (Henschke 1964, p. 467; Watson and Gilfillan 1976, p. 292; Buxton, Transactions RINA 1976, p. 316)

Approximation Formulae

Diesel Engines for Cargo Ships:

Watson–Gilfillan

$$W_{\mathrm{M}}[t] = C_{\mathrm{MD}} P_{\mathrm{B}}^{0.89}$$

where

P_{B} (kilowatt): break power of main engine
$C_{\mathrm{MD}} = 0.21$ (medium-speed diesel)
$= 0.30 \div 0.50$ (low-speed diesel)

Steam Turbines for Cargo Ships according to Buxton

$$W_{\mathrm{M}}[t] = C_{\mathrm{MT}} P_D^{\;0.5}$$

where

P_{D} (kilowatt): delivered power at the propeller
$C_{\mathrm{MT}} = 10.2$ single-screw
$= 14.1$ twin-screw
$= 5.8$ small ships

The above-introduced coefficients C_{MD} and C_{MT} can be actually adjusted to the particularities of the subject ship, based on the data of similar machinery installations.

Typical values for the machinery weight of slow-speed and medium-speed diesel engines are:

$$100-140\,\text{kp/kW, for powers of } 3{,}000-20{,}000\,\text{kW},$$

while the average value for turbocharged diesel installations of power 3,000–12,000 kW is: 130 kp/kW.

D. **Approximation based on the weight of main engine**

The basic reasoning of this method is similar to that of the previous section. On condition that there are comparative data from other ships with similar machinery installations, the calculation of the W_{M} weight is reduced to the weight of the main engine (plus gear unit, if any), which can be calculated accurately from the manufacturers' lists, especially for diesel-engine ships.

According to Watson and Gilfillan, the total weight of diesel-engine installations can be approximated as follows:

$$W_{\mathrm{M}} = W_{\mathrm{MM}} + W_{\mathrm{MREST}}, \text{ where}$$

$$W_{\mathrm{MM}} = 12\left(MCR_1 / RPM_1 + MCR_2 / RPM_2 + \ldots + MCR_N / RPM_N\right), \text{where}$$

MCR_{i}: MCR of engine (i), RPM_{i}: revolutions per minute of engine (i), N: number of engines

$$W_{\mathrm{MREST}} = C_{\mathrm{m}} \left(MCR\right)^{0.70}, \text{where}$$

$$\begin{aligned} C_{\mathrm{m}} &= 0.69\,(\text{bulkcarriers, cargo, and containerships}) \\ &= 0.72\,(\text{tankers}) \\ &= 0.83\,(\text{passenger ships and ferries}) \\ &= 0.19\,(\text{frigates and corvettes, for MCR in kilowatt}) \end{aligned}$$

Typical values of specific weights (kp/kW) of marine diesel engines are given in the following; note, however, that they do not include the weight of lubricants and cooling water:

Slow-speed diesel (110–140 RPM)	35–46 kp/kW
Medium-speed diesel (400–500 RPM in series)	15–20 kp/kW
Medium-speed diesel (400–500 RPM V type)	11–15 kp/kW
High-speed diesel (1,000–2,600 RPM)—large ones	≥4 kp/kW

For directly driven diesel-engine installations (low-speed diesel), W_M weight can also be calculated as follows (Schneekluth 1985):

$$W_M = C_{M1} W_{MM}$$

where

W_{MM}: main engine weight (tonnes)

$$C_{M1} = 2.2 \div 3.6, \text{ average value} = 2.6.$$

The coefficient C_{M1} can be adjusted to the under design ship based on comparable data of a parent ship.

For indirect diesel engine installations (medium-speed diesel with gear units) it applies correspondingly:

$$W_M = C_{M2}\left(W_{MM} + W_{MG}\right)$$

where W_{MG}: weight of gearbox, including clutch (tonnes)

$$C_{M2} = 3.5\left(\text{upper limit of } C_{M1}\right).$$

The weight of the gearbox (and clutch) can be calculated based on the manufacturers' catalogues and is a function of the main engine's power, the developed thrust of the ship, input/output revolutions per minute, and the construction method (layout of gears, way of housing—cast or welded).

For gearboxes with welded housing and 100 RPM exit speed (to the propeller), the specific weight is 3–5 kp/kW, while for a casted housing the weight is up to three times higher (see Henschke 1964, pp. 87–93).

For propeller speeds n_P larger than 100 RPM, but within the typical limits of merchant ships, the specific weight of the gearbox in (kp/kW) is about: is about:

$$(3.4 \text{ to } 4.0) \cdot 100 / n_P \,(\text{RPM})$$

Nevertheless, calculating the weight of the gearbox and that of the required clutch separately, if the latter is not integrated in the gear unit, the specific weight increases by the factor two (see Ehmsen, HdW XII, p. 250 (Handbuch der Werften, Schiffahrtsverlag-Verlag HANSA, Hamburg) and Volume 2 in Papanikolaou 2009a).

For turbine ships, indicative values for the specific weight of the total engine plant range between 15 ~ 19 kp/kW. This weight includes: steam turbines, gearboxes, boilers with water, and condensers. Its analysis shows that about 50 % of this weight refers to the weight of the boilers filled with water.

E. Calculation based on the breakdown of the W_M into subgroups

This method combines the use of accurate individual weights, if they can be calculated, and the use of empirical coefficients for the more complex subgroups.

H. Schneekluth (1985) proposed to analyze the machinery weight by dividing it into four subgroups:

I. Engine installation.
II. Electrical generator units.
III. Other weights except I & II
IV. Specific weights for ships of *special* mission

I. Engine installation

I1. Main engine: from manufacturers' catalogue
I2. Gearbox–clutch: from manufacturers' catalogue
I3. Shaft (without bearings)
a. Diameter propeller shaft end: According to the regulations of recognized classification societies, for instance, according to GL, for materials (like propeller shaft's higher tensile steel) with a tensile strength 700 N/mm^2 the following is concluded:

$$d_S(\mathrm{m}) = 11.5\left(P_D / n_P\right)^{1/3}$$

where

P_D (kilowatt): delivered power at the propeller
n_P (RPM): propeller revolutions per minute

b. Weight/length of shaft:

$$W_{SH} / l_{SH} \approx 0.081\left(P_D / n_P\right)^{2/3}$$

where

l_{SH}: length of shaft

I4. Propeller (s)

The weight of ordinary manganese bronze propellers may be estimated by:

$$W_{\mathrm{PR}} = K_{\mathrm{P}} D_{\mathrm{P}}^{3}\,(\mathrm{t})$$

where

D_{p} (meter): diameter of propeller

$$K_{\mathrm{P}} \cong \frac{d_{\mathrm{S}}}{D_{\mathrm{P}}}\left(1.85\frac{A_{\mathrm{E}}}{A_{\mathrm{O}}} - \frac{z-2}{10}\right)$$

This holds for fixed-pitch propellers with z blades and areas ($A_{\mathrm{E}}/A_{\mathrm{O}}$) (according to Schneekluth 1985) and

$$\begin{aligned} K_{\mathrm{P}} &= 0.12-0.14, \text{controllable-pitch CP propellers (merchant ships)} \\ &\quad\ 0.21-0.25, \text{controllable-pitch CP propellers (naval ships).} \end{aligned}$$

Alternatively, according to E. Strohbusch (1971), for fixed-pitch propellers:

$$W_{\mathrm{PR}} = D_{\mathrm{P}}^{2} \cdot d_{\mathrm{S}}\,(A_{\mathrm{E}}/A_{\mathrm{O}} + 0.2) \cdot K_{\mathrm{P}}'\,(\mathrm{t})$$

where $K_{\mathrm{P}}' = 1.2$–1.3 for manganese bronze propellers

II. Electric generator units (Schreiber, Journal Hansa 1977, p. 2117)

The approximation of the weight of the electric generator units (*gen-sets*) can only be done if we know the required electrical energy and the units' total power.

The electrical energy balance of a ship, which leads to the estimation of the required powering supply for electricity, must be done for the following operating conditions of the ship:

1. Sailing at design speed, en route
2. Course on alert/maneuvering, limited waters
3. Loading and unloading with own means
4. Immobilization (docking)

Usually for a commercial cargo ship the condition (2) is the most crucial in terms of requirements for electricity power.

Based on the required electrical power/energy, where all losses as well as the extent of simultaneous use of the various energy consumers should be included, the required power of the electric generators can be estimated.

The weight of the electric generators installation is a function of the way electricity is being generated:

- (1) Connection of the electrical generator(s) through a gearbox to the propulsion machinery (*shaft generator*). This may cover parts of the electrical energy needs.
- (2) Diesel-engine powered generator set by use of medium-speed/high-speed diesel engines.

In the second case, the weight of the diesel engine/generator unit (gen-set) may be approximated by:

$$W_{\mathrm{EP}}/P = 15 + P/70\,(\mathrm{kp/kW})$$

where P (kilowatt): power of the individual generator set.

In case of (1) significantly smaller weights are concluded, because of the higher efficiency of the main diesel engines. However, this option requires the existence of controllable-pitch propellers so that the speed/revolutions of the propulsion engine driving the electrical generator can be kept constant, when slowing down the ship; on the other hand, for the standby/maneuvering/anchoring mode, when approaching to the port or in case of emergency, it must be switched to an independent electric generator unit (2), but to a limited extent.

III. Other weights

This category includes all the weights of the machinery installation that were not mentioned in I and II, that is, pumps, pipes, boilers, exhaust absorbers, cables, splitters, spare parts, ladders, gratings, day tanks, gas containers, condensers, separators, oil coolers, water cooling system, engine room control system, noise, and thermal insulation of the engine room.

$$\text{Typical values:}\quad W_{\mathrm{III}}/P_{\mathrm{B}} = 40 - 70\,(\mathrm{kp/kW})$$

where the lower limit applies to large installations of over 10,000 kW, as a function of the engine room volume.

IV. Specific weights (only for certain ship types)

- a. *Tankers*
  - Cargo pumps and pipes
  - Steam generating boilers (heating of cargo, tank cleaning) 120–180 kp/kW
- *b. Reefers*
  - Cooling system (without air ducts): weight per net refrigerated volume (Net–Net) 14 kp/m^3
- *c. Refrigerated cargo containerships*
  - Refrigeration facilities: indirect cooling 1 t/FEU container; direct cooling 0.7 t/FEU container

– Ducts of chilled air: indirect cooling 0.8 t/FEU container; direct cooling 1.3 t/FEU container

Additionally the weight of the thermal insulation of reefer cargo is mentioned, though it belongs to the W_{OT} group.

- 50–60 kp/m^3 volume net–net (refrigerated cargo)
- 1.9 t/FEU container (bananas; containership)
- 1.8 t/FEU container (meat; containership)

2.15.7 Analysis of Deadweight DWT

In the case of cargo ships, the owner usually predefines/specifies the total deadweight DWT, rarely the payload weight W_{LO}. However, independently of the knowledge of the total value of DWT in the initial design phase, this DWT value must be broken down into its components and be carefully analyzed. This enables a better estimation of the mass centers of the various DWT components and of the influence of individual weight elements, which constitute the DWT, on the arrangement of spaces of the vessel (e.g., tank spaces for fuel, ballast, etc.) and on the overall ship design and performance.

It is estimated that the deadweight of a ship *decreases* with the increase of the ship's age, namely by approximately 5 ‰ in the first year and by 0.5 ‰ over the next years, *due to the increase* of the light-ship weight W_L. Typical reasons for the increase of W_L are: paintworks, corrosion of plates, added spare and reserve equipment, and waste and residues of liquids, especially in the bilges and other waste tanks.

DWT has already been defined in Sect. 2.4.1 as follows:

$$DWT = W_{LO} + W_F + W_{PR} + W_P + W_{CR} + B$$

Payload

The payload may be defined as the difference:

$$W_{LO} = DWT - \left(W_F + W_{PR} + W_P + W_{CR} + B\right)$$

where the individual weights W_F, W_{PR}, W_P, W_{CR}, and B are calculated in the following:

Weight of fuels W_F (includes also the weight of lubricants)

The required fuel is calculated for a round trip from/to the departure/replenishment port (without refueling), unless the owner specifies this differently. The required fuel can be approximated by the following formula:

$$W_{\mathrm{F1}} = \left(P_{\mathrm{B,1}} \cdot b_1 \cdot t_1 + P_{\mathrm{B,2}} \cdot b_2 \cdot t_2 / \eta_{\mathrm{E}}\right) \cdot C \cdot 10^{-6}$$

where

W_{F1}: weight of fuel (tonnes)
$P_{B,1}$: required power of main engine (depending on speed and operating conditions) (kilowatt)
$P_{B,2}$: average required power of electrical generators (kilowatt)
t_1: time of a roundtrip voyage (hours) based on the service speed and operating range = range(sm)/service speed(knots)
t_2: operating time of electric generators (hours) = t1 + time at port
b_1: specific consumption of the main engine (gram per kilowatt-hour)
b_2: specific consumption of auxiliary engines for electric generators (gram per kilowatt-hour)
η_E: average efficiency of electric generator units
Margin reserve: $C \equiv 1.2–1.4$

The constant *C* refers to the reserve for overconsumption due to change of course, unpredictable waiting, assistance to other ships in case of emergency, and residues in the tanks (2–4 %).

It is assumed that the influence of the sea state, winds, and hull fouling on fuel consumption has been already accounted for during the estimation of the *service speed* and the corresponding required propulsion power.

The specific weight of fuel and lubricant oils varies significantly, depending on their quality and use.

On average we have:

Marine diesel oil (*MDO* fuel)	0.85 t/m^3
Heavy fuel oil for slow-speed diesel engines and boilers (*HFO* fuel)	0.92–1.02 t/m^3
Lubricant oil	0.928 t/m^3

For cargo ships it may be considered, as to the consumption of auxiliary engines, that this corresponds to 5–7 % of the required fuel for the propulsion engine.

In addition to the above consumptions, the corresponding values for heating must be added, if it was not included in the consumption of the auxiliary machines (central heating) or the heating is provided by exploitation of the engine's exhaust gas' high temperature. Likewise, for tankers the production of steam for cleaning/heating of the cargo tanks should be added.

The specific consumptions for different types of main engine installations are shown in Fig. 2.92, as a function of the type of main engine's type (diesel of slow- and medium-speed, steam turbine, and gas turbine) and its loading rate. It is

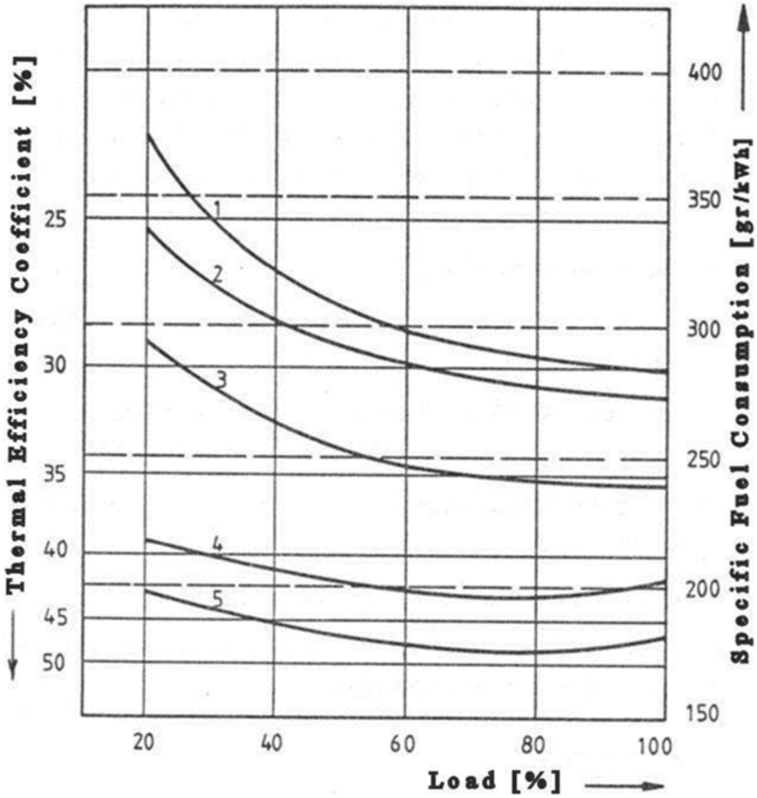


Fig. 2.92 Specific fuel consumption and thermal efficiency coefficient of marine engines. *1* gas turbine, *2* steam turbine 12 MW, *3* steam turbine 30 MW, *4* medium-speed diesel engine, *5* slow-speed diesel engine

evident that for diesel engines the minimum specific consumption corresponds to approximately 85 % of the MCR of the installed power, while for turbo engines the minimum consumption corresponds to 100 % loading. Nevertheless, regardless of manufacturer, the specific consumption is absolutely minimal for low-speed diesels (~ 170 g/kWh; today down to about 155 g/kWh), followed by medium-speed diesels (~ 190 g/kWh; today down to 175 g/kWh), the steam turbines (290 ~ 330 g/kWh, today down to 250 g/kWh, depending on the power magnitude, the loading, the type and manufacturer) and finally, the gas turbines (300 ~ 350 g/kWh, today down to 270 g/kWh). It should be noted that beyond the specific fuel oil consumption (SFOC), of interest for the *cost of fuel*[26] is the *type* of fuel consumed, with heavy fuel oil (HFO for low-speed diesel engines and steam turbine boilers) being the least expensive per ton fuel, followed by marine diesel oil (MDO, for medium- and

[26] Indicative Fuel Oil Prices (June 2014): Heave Fuel Oil (IFO380) Singapore: 617.50 USD/ton, Rotterdam: 602.50 USD/ton, Houston: 612 USD/ton, Marine Diesel Oil (MDO): 915.50 USD/ton.

high-speed turning diesel engines). Modern marine gas turbines can run a wide variety of fuels.

Weight of lubricants W_{F2}

This concerns the weight of lubrication oil. The consumption is:

Diesel engines: 0.15 gr/kWh circulation lubricant
0.6–1.4 gr/kWh cylinder lubricant

(note that medium-speed diesel engines without crosshead require cylinder lubricants *also* for the circulatory system).

Turbines and gearboxes:
0.1–0.2 g/kWh

The weight of the lubricants corresponds approximately to 3–5 % of the fuel weight (diesel engines) and is usually in the order of 20 t for medium-speed and 15 t for low-speed diesel engines. When carrying out an accurate calculation for the size of the related tanks for lubricants, based on the kilowatt-hour, it is recommended to take into account the consumption for about 50 journeys.

Water supplies

We distinguish the following types and qualities of onboard water:

- fresh water, drinking, and cleaning water,
- feeder water for boilers and cooling network,
- seawater for sanitary tanks, if fresh water is not used,
- ballast water

Typical values

Freshwater:

Drinking: 10–20 kg/person/day
Cleaning: 120 kg/person/day, if the accommodation has showers,200 kg/person/day, for accommodation with bath tubs.

Feeder for the boilers: 0.1 kg/kWh plus the liquid for filling the network.

The water supplies of a ship are usually not sufficient for the entire duration of a voyage. The needs are partly covered through the refilling at intermediate ports or through the production of fresh water with onboard seawater desalination plants.

Contemporary desalination equipment aboard ships allows freshwater production from seawater using either a *thermal* or a *membrane type* (*reverse osmosis*) desalination process. In the thermal distillation process the seawater evaporates and the vapor condenses thereafter producing clean freshwater. More efficiently, evaporation is conducted at low pressure so that the heat of the engine's cooling water can be used for the heating process. Particularly, evaporation of seawater at 40 °C occurs at 93 % vacuum. Thus, the cooling water of the main engine (with exit

temperature of about 32 °C) requires a little reheating by about 8 °C to be used for the desalination process. It is estimated that with 1 kg oil for the additional heating, it is possible to produce this way 100 kg of freshwater (Schneekluth 1985). Nowadays, multistage evaporators are commonly used aboard passenger ships, with increased needs for fresh water, whereas *tube bundle evaporators* are prevailing aboard cargo ships. (see Meier-Peter and Bernhardt 2009).

As for the drinking water, the requirements in terms of quality are nowadays enhanced so that the refilling from ports with adequate sanitary conditions is preferred.

Note that for a standard *cargo ship* the amount of carried fresh water is in the range of 80–100 t; however, the needs of *passenger ships,* particularly of cruise ships, are much higher. Depending on the size and type of ship, desalination plants of production capacity between 5 and up to 100 t water per day are installed onboard modern ships, with the large passenger ships standing on the top of consumers.

Weight of supplies—food

The weight of supplies/food is estimated by roughly: 7–16 kg/man/day. This weight concerns not only daily consumption, but also the reserve for delays of voyage, deterioration of food, and delays of supply.

Weight of passengers and luggage

Passengers:	75 kg/passenger
Luggage:	20 kg/passenger, for short trips60 kg/passenger, for long voyages; holds also for crew members.

Weight of ballast water

It should be considered that for a well-designed cargo ship, in the design load condition[27], ballast water should not be necessary. The carriage of ballast water negatively affects the ship's economy both with respect to the additional carried weight (at the expense of not carried payload), the associated fuel cost and the cost of ballast water treatment (see, IMO Res. MEPC. 173(58), 2008b).

Typical reasons that lead a designer to the planning of ballast are:

- insufficient stability, especially after the consumption of fuel/supplies (end of voyage)
- balancing of trim, especially for ships with the engine room abaft
- to increase the draft at bow/stern and avoid slamming and propeller racing phenomena

[27] Exceptions to the rule are the containerships, especially when in the full load/design condition they are expected to carry many containers on deck (causing a high center of ship's mass). This leads to a significant amount of ballast in the full load/design condition, to ensure adequate GM; consequently, for a given DWT, the overall payload capacity decreases. Recent containership design developments and ship design optimizations/innovations, however, look for minimum ballast (*zero ballast* ships).

- to smoothen the longitudinal stresses due to uneven cargo hold loading (e.g. ore carriers, containerships)
- to avoid kipping and dumping during ship launching

In addition, the international regulations of MARPOL specify for tankers over 20,000 t DWT that the trim is limited, namely $\delta T < 0.015\ L_{PP}$ for the ballast condition.

The distribution of adequate ballast tank space along the ship and the provision of sufficient amount of ballast water results from the requirements of the extreme ballast condition.

If we assume that in ballast condition it is required that we have:

- abaft: full immersion of propeller
- forward: $T \geq 0.02\ L_{PP}$

then it is concluded for the ballast water weight:

$$W_B = \Delta_B - \left(DWT_R + W_L\right)$$

where

W_B: ballast water weight
Δ_B: displacement in ballast condition
DWT_R: sum of rest fuel, rest payload, remaining supplies and weight of crew with luggage
W_L: light ship weight

The desired average draft in ballast condition is:

$$T_B = \left(D_P + e + 0.02L\right) / 2$$

where

D_P: propeller diameter
e: distance of lower extremity of the propeller blades from the base.

The displacement in ballast condition is:

$$\Delta_B = w_{SW} \cdot L \cdot B \cdot T_B \cdot C_{BB}$$

where

w_{SW}: specific weight of seawater
C_{BB}: block coefficient in ballast condition $= C_B - C\left[\left(T - T_B\right) / T\right] \cdot \left(1 - C_B\right)$

where

C_B: block coefficient for design draft

T: design draft
C: constant ~0.4.

Thus, the minimum amount of carried ballast in the ballast load condition is this way estimated and helps the designer to plan for sufficient tank space and arrangements of ballast tanks.

Permanent ballast

Permanent ballast is required for certain types of ships, for example, sailing boats, and often for *converted* ships[28], with stability problems. This ballast weight is generally not included in the DWT, but in the weight of the steel structure (eventually the ship's light-ship weight). Marine concrete is often used as permanent ballast material because of its low cost. It is mainly placed on the ceiling of the double bottom; a specific marine concrete ballast weight of about 4 t/m^3 can be achieved, whereas with the use of heavy $BaSO_4$ (barium sulfate oxide) the specific weight can reach values of 4.6 t/m^3. In some converted RoPax ships, permanent ballast can also carried in the form sea water, which is placed in *permanent ballast tanks;* the latter are "sealed" by the authorities to avoid stability problems by improper use during operation.

2.16 Verification of Displacement

Based on the approximations of the individual weight components of the ship (Sect. 2.15) and given her deadweight DWT (for ordinary cargo ships), the total weight of the ship under consideration is expressed as:

$$\Delta = W_L + DWT$$

where

W_L: light-ship weight

$$W_L = W_{ST} + W_{OT} + W_M + R$$

W_{ST}: weight of steel structure
W_{OT}: weight of outfitting

[28] In the past and in many countries around the world, it was popular to covert cargo ships (mainly general cargo type of ships) into passenger ships (mainly RoPax ships) by keeping the main hull unchanged. Trivially, with the added high superstructures typical to passenger ships, the stability of these ships could only be kept within regulatory margins by adding permanent ballast. In many cases this was accompanied by more severe design measures, like the fitting of streamlined "sponsons" on the ship's hull, increasing the ship's breadth and form stability. The latter design measure was also applied independently of the carried permanent ballast.

W_M: weight of machinery and propulsion plant
R: margin—tolerance

and

DWT: $DWT = W_{LO} + W_F + W_{PR} + W_P + W_{CR} + B$
W_{LO}: payload weight
W_F: fuel/lubricant weight
W_{PR}: weight of provisions and water
W_P: weight of passengers and their baggage
W_{CR}: weight of crew and their baggage
B: weight of *nonpermanent ballast* (water), for a specified draught and satisfactory stability and trim.

The comparison of the sum of the weight components, namely Δ, with the weight of water displaced by the vessel's hull shows to what extent the approximations of the weight components are in line with the designed hull.

$$\Delta = w_{SW} \cdot \nabla'$$

where

w_{SW}: specific weight of sea water
$\cong 1.025$ t/m^3 (mean value)
Δ': corrected moulded hull volume
$= C_B \cdot L_{PP} \cdot B \cdot T \cdot K$
K: *moulded hull correction coefficient,* accounting for an average thickness of the ship's outer shell plating
$= 1.0035$ for tankers
$= 1.005$ for cargo ships
$= 1.006$ for shortsea cargo ships
$= 1.007$ for containerships

If the difference between Δ (the ship's weight) and $W_{SW} \cdot \nabla'$ (weight of displaced water) is within the margin *R* of W_L, the design can proceed to the next phase. Otherwise, if the Δ weight exceeds the displacement more than the *R,* the hull must be modified accordingly to balance the difference. The margin of tolerance of *R* varies (see Sect. 2.15.1) between 1~3% W_L, in the preliminary design phase, while according to other sources (Schneekluth) it could reach 6% W_L (for more complicated ships).

2.17 Verification of Holds' Capacity

2.17.1 Definitions

a. *Gross volume* (German: Bruttoladeraum) ∇_G: Corresponds to the holds' volume bounded by the *outer edge* of the holds' frames, of the deck beams or the *inside*

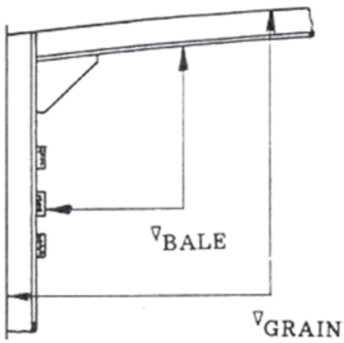


Fig. 2.93 Holds' volume for bulk (grain) and bale cargo $\nabla_{GR} \approx (0.990 \div 0.995)\ \nabla_G$, $\nabla_B \approx (0.90 \div 0.93)\ \nabla_{GR}$

edge of the shell plating, of double bottom, of bulkheads and of the ceiling deck; it includes the volume occupied by the frames and strengthenings or other structural fittings that is not deducted[29]

Dimension units:	(m^3) or (ft^3), 1 $m^3 = 35.32\ ft^3$
Symbol/relationships:	$\nabla_G \approx \nabla_S$ (∇_S: molded hold volume as calculated by integration of sectional areas)

b. *Grain volume* (German: Kornladeraum) ∇_{GR}: Corresponds to the volume that grain (or liquid) cargo occupies when filling the hold, that is, it is equal to the gross volume defined in (a) subtracting the volume of strengthenings and other fittings (e.g. holds' planking)(see Fig.2.93).

c. *Bale volume* (German: Stückgutvolumen) ∇_B: Corresponds to the holds' volume that is bounded by the *inside edge* of the plating of the double bottom or its planking, the inside edge of the deck beams, the inside edge of the side strengthenings of the hold or section and finally the inside edge of the side planking or the bulkheads' strengthenings.

Units:	(cubic meter) or (cubic feet)
Symbol/relationships:	$\nabla_B \approx (0.90 \div 0.93)\ \nabla_{GR}$ (lower limit: for sharp/slender ships)

d. *Net hold volume (reefer ships)* (German Netto-Volumen) ∇_N: It refers to the holds' volume for *refrigerated* cargo and is bounded by the inside edge of the insulation planking of the hold space.

Units:	(cubic feet) or (cubic meter)
Symbol:	∇_N

[29] This volume corresponds to the holds' volume resulting from the ship capacity curves, thus by integration of the areas of the sections belonging to and bounding the respective hold (see Sect. 2.17.2).

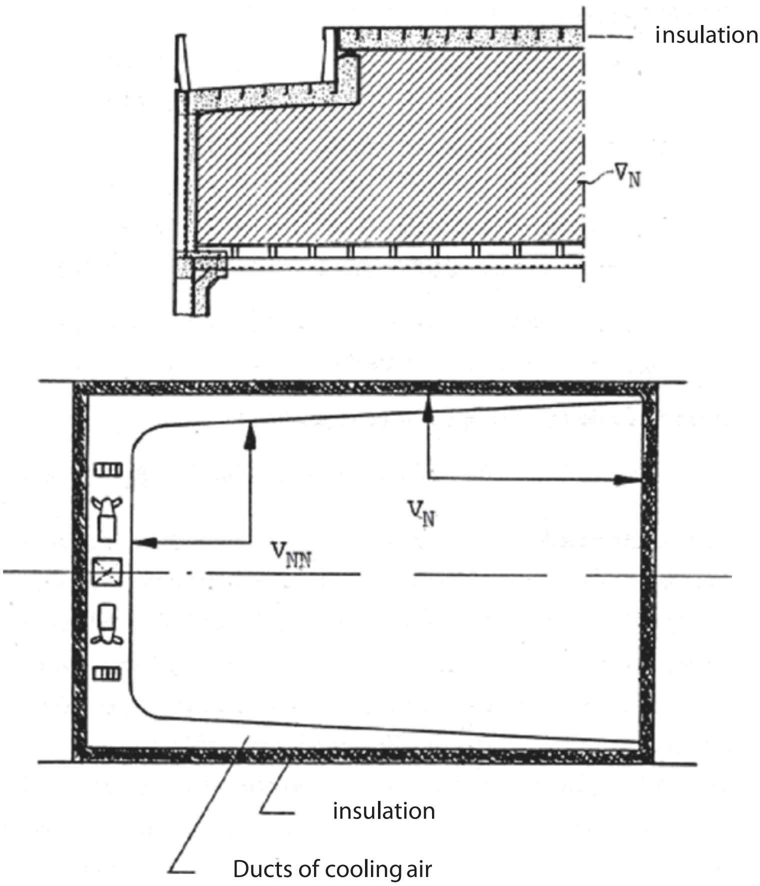


Fig. 2.94 Holds' net volumes for refrigerated cargo

e. *Net–net volume (reefer ships)* (German: Netto-Netto-Volumen) ∇_{NN}: It corresponds to the net volume defined in (d) subtracting the volume occupied by the refrigeration facilities (e.g., ducts of cooling air, coolers, etc.) (see Fig. 2.94).

Units: (cubic feet) or (cubic meter)

Symbol/relationships: $\nabla_{NN} \cong (0.60\text{–}0.63)\,\nabla_S$ (horizontal ventilation)

$\cong (0.65\text{–}0.69)\,\nabla_S$ (vertical ventilation)

f. *Capacity coefficient* (German: Räumte) and *stowage factor*: The capacity coefficient is defined as the ratio of the holds' volume (usually bulk-grain volume) to the deadweight of the ship

$$R = \nabla_{GR} / DWT.$$

The *capacity coefficient* is an *attribute of the ship*. Instead, the ***stowage factor*** (*SF*), which corresponds to the required hold volume per ton of cargo, is an *attribute of the cargo*.

Units: (cubic meter per ton) or (cubic feet per ton)

Examples (Capacity factor)

General cargo ship	1.6–2.0 m^3/t
	(55–72 ft^3/t)
Small–medium tanker	1.3–1.4 m^3/t
(< 100,000 t DWT)	(45–49 ft^3/t)
Large tanker	1.2–1.25 m^3/t
(> 100,000 t DWT)	(43–44 ft^3/t)

Examples (SF)
Light cargoes $SF \geq 2.0\ m^3/t$

Citrus and other fruits	2.0–2.5 m^3/t
Cotton goods	2.2–2.8 m^3/t
Coking coal	1.95–2.78 m^3/t
Tobacco	3.00–5.00 m^3/t
Bananas (in boxes)	3.25 m^3/t

Semiheavy cargoes $1.25 \leq SF \leq 2.0\ m^3/t$

Grains	1.2–1.8 m^3/t
Sugar (in sacks)	1.29–1.34 m^3/t
Coal	1.18–1.33 m^3/t
Coffee	1.61–1.75 m^3/t
Wines	1.39–1.53 m^3/t

Heavy cargoes $SF \leq 1.25\ m^3/t$

Cements	0.64–0.78 m^3/t
Ores	0.34–0.50 m^3/t
Crude oil	0.91–1.00 m^3/t
Steel panels	0.60 m^3/t
Electrical cables	0.85–1.12 m^3/t

g. *Gross tonnage* (German: Bruttoraum): It is the result of application of relevant national and international *tonnage measurement regulations* and forms an important information element regarding the *size (total enclosed volume)* of the measured ship. This value corresponds to the enclosed volume of all closed spaces of the ship (that is, not only of the holds), without this correspondence to be mathematically conclusive, due to the *exclusions* of certain spaces (e.g., fore/aft peak tanks, ballast tanks, wheelhouse, galleys, and public areas). The gross

tonnage forms in general a reference baseline for determining the number and composition of the crew, the implementation of safety regulations, for determining the ship's classing fees, as well as other costs (taxes, insurance, transit fees of canals, etc.).

Units: GRT (gross register tons) or GT (gross tonnage), 1RT $=100\ \text{ft}^3=2.832\ \text{m}^3$

h. *Net tonnage* (German: Nettoraum): Like the gross tonnage defined in (g), the *net tonnage* is the result of application of relevant *tonnage measurement regulations* and is a representative quantity for the "*economic value*" (commercial exploitability) of the ship. The net tonnage is calculated from the gross tonnage, which is reduced by some "*deductible*" spaces that are not exploitable for cargo transport (e.g., the machinery space, spaces of pump rooms/auxiliary machinery, and crew accommodation) . *The net capacity cannot be smaller than 30 % of the gross tonnage*. The magnitude of the net tonnage/capacity is used, like that of the gross tonnage, to calculate various fees, for instance, port charges, etc.

Units: NRT (net register tons) or net tonnage (NT)

The international regulations of tonnage measurement of ships (International Tonnage Measurements of Ships) may be found on IMO's website (http://www.imo.org/Conventions); they apply to all ships longer than 24 m and built after 18 July 1982. In accordance with these regulations, the following relationships between the ship's tonnage and the ship's main characteristics apply:

Gross Tonnage (GT)

$$GT = \left(0.2 + 0.02\log_{10} V\right)V$$

where V is the volume of all the enclosed spaces of the ship.

Net Tonnage (NT)

$$NT = K_2 V_c \left(\frac{4d}{3D}\right)^2 + K_3\left(N_1 + \frac{N_2}{10}\right)$$

where

(a) The coefficient $\left(\frac{4d}{3D}\right)^2$ should not be larger than 1.0.

(b) The coefficient $K_2 V_c \left(\frac{4d}{3D}\right)^2$ must not be smaller than 0.25 GT.

(c) The net tonnage NT must be greater than 0.30 GT.

V_c = total volume of holds' space (cubic meter)
K_2 = $0.2+0.02$ log10 V_c

$K_3 \quad = 1.25\dfrac{GT+1000}{1000}$
$d \quad$ = draught[30] at amidships
$D \quad$ = side depth to the uppermost deck at amidships
$N_1 \quad$ = number of passengers in cabins with more than eight passengers
$N_2 \quad$ = number of the remaining passengers
$N_1+N_2 \quad$ = total number of passengers that the ship can carry in accordance with her safety certificate. For passenger numbers N_1+N_2 less than 13, thus in case of cargo ships, then the N_1 and N_2 are set to zero.

It is obvious, that the "physical capacities of the holds" defined by the volumes (a) ∇_G, (b) ∇_{GR} and (c) ∇_B have nothing to do with the *tonnage capacities* determined in accordance with the tonnage regulations defined in (g) and (h).

Beware of *nonscientific* literatures/references:

Ship *capacity* or *tonnage* of 1 t usually means:

- tankers, bulkcarriers: t DWT
- ROPAX/cruise ships: GRT
- general cargo ships: t DWT, rarely GRT
- warships: tons Δ (displacement).

2.17.2 Calculation of Hold Volume

A. **Volumetric/capacity curves**

Provided that there is at least a preliminary shiplines plan (or sketch) of the subject ship, then the calculation of the hold volume and the volumetric distribution of spaces can be derived through the "volumetric/capacity curves."

The *volumetric/capacity curves plan* (German: Raumkurvenblatt), are drawn with the same ordinates as the corresponding curves of sectional area lengthwise for the various draughts concerned, but herein at the level of double bottom, of intermediate deck positions and of the uppermost deck (see Fig. 2.95).

In the *volumetric/capacity curves plan,* which resembles the sectional area curves plan, the boundaries of the various hold spaces are also sketched, for example, the deck and bulkhead boundaries; furthermore, there is information about the usability of the spaces and the corresponding exploitable volume (see example).

The areas below the volumetric/capacity curves correspond to the volume of the indicated spaces; volume numbers can be easily obtained by integration of the areas using Simpson's rule or a mechanical *planimeter* (in old times). The concept of capacity curves can be successfully used both in the initial design, and in advanced stages, if the shiplines are available. The volumetric/capacity curves plan is also useful for a rapid assessment of the longitudinal position of the center of DWT

[30] Summer draught or subdivision draught (RoPax ships) amidships.

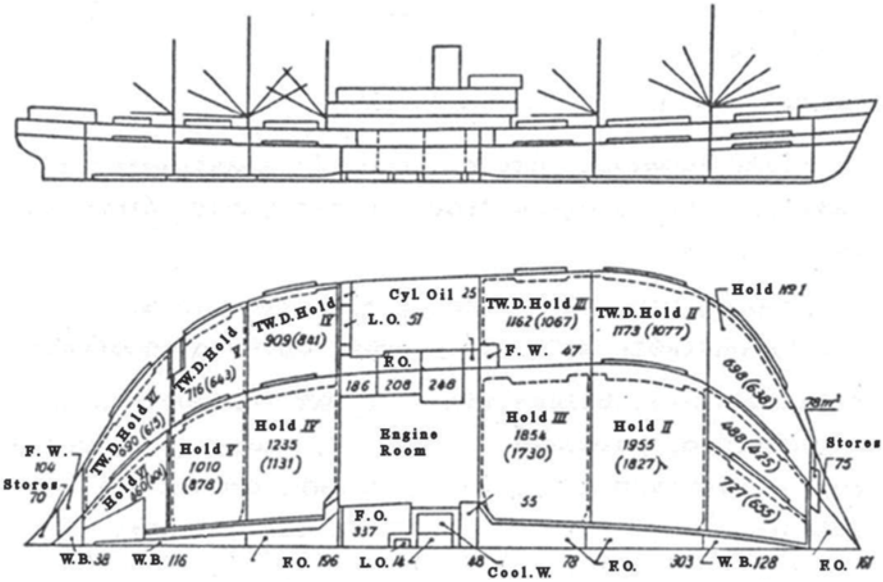


Fig. 2.95 Volumetric/capacity curves plan of a cargo ship. The *dashed lines* in the plan indicate the available space for bale cargo (∇_{BALE}). The *numbers* (in cubic meter) show the available volume for bulk cargo (∇_{GRAIN}), and in *parentheses* the volume of bale cargo (∇_{BALE})

through a moment balance of the longitudinal moments of the various DWT weight components.

B. **Below main deck volume** (initial design phase)

The volume below the main deck forms the basis for the approximation of not only the hold volume, but also of the engine room spaces, of double bottom and of other tanks. Furthermore, it is a necessary element for calculating the ship's steel weight (see Sect. 2.15.4, B3).

If during the early design phase (feasibility), the preliminary shiplines (or hydrostatic diagrams) are not available, the following simplified procedures are proposed to approximate the volumes.

B.1. **Cubic coefficient method**

Provided that there are sufficient comparative data from one or more similar ships it is possible to define the *cubic* coefficient:

$$C_{\nabla_C} = \frac{(\nabla_C)_0}{L_0 \cdot B_0 \cdot D_0}$$

where

∇_C: comparable hold volume (grain or bale),
L, B, D length, beam, and side depth,
$(\,)_0$: parent ship.

Thus, it is concluded for the subject ship's (index 1) hold volume:

$$(\nabla_C)_1 = C_{\nabla_C} \cdot L_1 \cdot B_1 \cdot D_1$$

If additionally a side view of the parent ship is available, from which it is possible to identify the *total length of the cargo hold spaces* L_C, then the above coefficient is better defined with respect to this length, that is:

$$C'_{\nabla_C} = \frac{(\nabla_C)_0}{L_{C0} \cdot B_0 \cdot D_0}$$

and

$$(\nabla_C)_1 = C'_{\nabla_C} \cdot L_{C1} \cdot B_1 \cdot D_1$$

where

L_{C1}: total length of hold spaces of the subject, under design ship.

B.2. **Method of circumscribed parallelepiped** (mainly applies to cargo ships with engine abaft)

Gross hold volume

$$\nabla_G \cong L_C \cdot B \cdot \left(D_S - h_{Db}\right) \cdot C_{BLC} + \nabla_H$$

where

∇_G: Gross hold volume defined in Sect. 2.17.1.a.,
L_C: overall length of cargo hold spaces,
B: beam of ship,
D_S: raised/corrected side depth for sheer/camber of deck,

$$D_S \cong D + 0.08 \cdot \left(S_F + S_A\right)$$

h_{Db}: average height of double bottom (including possible planking)
C_{BLC}: Local fullness coefficient of hold volume,

$$\cong (C_B + 2)/3 \text{ or}$$
$$\cong (C_{BD} + 1.05)/2$$

∇_H: volume of hatchways.

B.3. **Approximation method of Schneekluth**

The method refers to the total volume *below the main deck* and has been earlier applied in calculating the steel weight (see Sect. 2.15.4, B.3).

The basic formula for the total volume *below the main deck*, ∇_{UD}, is expressed as:

$$\nabla_{\mathrm{UD}} = \nabla_{\mathrm{D}} + \nabla_{\mathrm{S}} + \nabla_{\mathrm{B}} + \nabla_{\mathrm{H}}$$

where

$$\nabla_{\mathrm{D}} = L \cdot B \cdot D \cdot C_{\mathrm{BD}} \text{ (volume up to the height of } D\text{)}$$

where

$$C_{\mathrm{BD}} = C_{\mathrm{B}} + C_4 \left[\left(D - T \right) / T \right] \cdot \left(1 - C_{\mathrm{B}} \right)$$

and

$$C_4 \cong 0.25 \text{ for hulls with small flare above waterline}$$
$$\cong 0.40 \text{ for hulls with large flare above waterline}$$

$\nabla_{\mathrm{S}} = L \cdot B \cdot (S_{\mathrm{F}} + S_{\mathrm{A}})/6$ (volume between D and a deck line accounting for longitudinal sheer of the deck, as applicable);

alternatively,

$$\nabla_{\mathrm{S}} = L_{\mathrm{S}} \cdot \mathrm{B} \cdot \left(S_{\mathrm{F}} + S_{\mathrm{A}} \right) \cdot C_2$$

where

$$C_2 \cong C_{\mathrm{BD}}^{2/3} / 6 \cong 1/7$$

L_{S}: length of sheer

$\nabla_{\mathrm{B}} = L \cdot B \cdot b \cdot C_3$ (volume due to a *beamwise* deck camber)

where

b: height of camber ($\cong 0.02 \cdot B$)

$$C_3 \cong 0.5 \div 0.6$$

∇_{H}: hatchways' volume$\cong L_{\mathrm{H}} \cdot B_{\mathrm{H}} \cdot h_{\mathrm{H}}$
L_{H}: overall length of hatchways
B_{H}: overall width of hatchways
h_{H}: average height of hatchways' coamings.

Based on the volume below the main deck ∇_{UD}, the hold volume can be calculated as percentage of ∇_{UD}, namely:

$$\nabla_G \cong k \cdot \nabla_{UD}$$

where

$$k = 0.6 \div 0.77$$

The coefficient k must be verified based on data of similar ships.

B.4. **Approximation method based on displacement**

Again the total volume under the main deck ∇_{UD} is sought.

The requested volume ∇_{UD} is supposed to consist of two parts

$$\nabla_{UD} = \nabla + \nabla_{AW}$$

where

∇: displaced volume at design waterline
∇_{AW}: hull volume *between design waterline and main deck*

The latter term is calculated as:

$$\nabla_{AW} = L \cdot B (D - \mathrm{T}) \cdot (C_{WP} + C_{WPD}) / 2 + \nabla_S + \nabla_B$$

where

C_{WP}: waterplane area coefficient$\cong (1 + 2C_B)/3$ (for nonpronounced sections, see Sect. 2.9)
C_{WPD}: deck waterplane area coefficient ($\cong 1.0$)
∇_S: additional volume due to sheer profile (see B.3)
∇_B: additional volume due to hatchways (see B.3)

B.5. **Method of Carstens** (cargo ships, aft engine room, see Carstens 1964, Journal Schiff and Hafen, p. 619).

2.18 Verification of Stability and Trim

One of the most important steps in the preliminary ship design stage is the verification/control of the ship's stability (to a lesser degree of the ship's trim, except for special cases) for the ship under consideration.

In the initial design stage it is sufficient to examine the *intact*[31] stability for small inclination angles (*initial stability*), what is essentially the control of the adequacy

[31] *Intact* stability: the stability of the ship assuming her buoyant hull intact.

of the metacentric height $\overline{GM}$. The stability control is complemented in the next steps of the design by examining the ship's stability curves (*stability for large inclination angles*); the latter requires an accurate knowledge of the ship's hull geometry that is usually not available in the first stage of design. In later stages of ship design, the ship's *damage*[32] stability also needs to be verified/examined against set damage stability criteria. Detailed reviews on the ship's stability, on calculation methods of the ship's stability and the in force stability criteria are given in the listed references Lewis (Vol. I, 1988), Papanikolaou (1982), Rawson and Tupper (1994). In section 2.18.8 of this book, the intact stability criteria of IMO are elaborated, whereas, in Appendix E a review of developments of the ship's damage stability criteria is presented.

At the stage of initial design, it is recommended to apply simplified formulas or diagrams/charts for the assessment of the ship's initial stability. As we know the metacentric height is derived as the difference between the ship's *form* and *weight* stability:

$$\overline{GM} = \overline{KM} - \overline{KG}$$

where the vertical position of the mass center of the vessel $\overline{KG}$ may be considered as approximately known (see Sect. 2.15.2), while the vertical position of the (transverse) metacenter:

$$\overline{KM} = \overline{KB} + \overline{BM}$$

is calculated through the estimation of vertical position of the center of buoyancy $\overline{KB}$ and vertical distance of metacenter from the initial center of buoyancy $\overline{BM}$ (*transverse metacentric radius*). Both values, unless more accurate data of the hull are available, are usually approximated through semiempirical/mathematical formulas as a function of the already known main particulars and hull form coefficients of the ship, what is elaborated in the following.

2.18.1 Vertical Position of Buoyancy Center

Normand I:

$$\overline{KB} = \mathrm{T}\left(0.9 - 0.36C_{\mathrm{M}}\right)$$

Schneekluth:

$$\overline{KB} = T\left(0.9 - 0.3C_{\mathrm{M}} - 0.1C_{\mathrm{B}}\right)$$

[32] *Damage* stability: the stability of the ship in case of loss of her watertight integrity (LOWI).

Normand II:

$$\overline{KB} = T(5/6 - C_B / 3C_{WP})$$

The accuracy of the above formulas for ordinary ship hulls is in the range of 1 % of T (according to Schneekluth). The third expression (Normand II) requires the knowledge of the waterplane area coefficient C_{WP}, which is often not known in the initial phase, namely prior to fixing the character of the sections (U or V), thus may change easily.

2.18.2 Metacentric Radius

All known approximation formulas for the metacentric radius $\overline{BM} = \mathrm{I}_T / \nabla$, are based on the appropriate approximation of the transverse moment of inertia I_T of the waterplane. The transverse moment of inertia can be easily deduced from the moment of inertia of the waterplane of the circumventing parallelogram, having the same length and beam like the ship's waterplane; thus, considering that transverse moment of inertia of the circumventing parallelogram is $L \cdot B^3/12$, we may correct it to account for the actual form of the waterplane, as expressed by the correction coefficient $C_1 = f(C_{WP})$. Thereby the following expression is concluded:

$$\overline{BM} = \frac{\mathrm{I}_T}{\nabla} = C_1 \cdot \frac{L \cdot B^3 / 12}{L \cdot B \cdot T \cdot C_B} = C_1 \cdot \frac{B^2}{12 \cdot T \cdot C_B}$$

where the correction coefficient $C_1 = f(C_{WP})$ is calculated as follows:
Normand

$$C_1 = 0.096 + 0.89 \cdot C_{WP}^2$$

Schneekluth

$$C_1 = C_{WP}^{1.8}$$

Bauer

$$C_1 = 0.0372(2C_{WP} + 1)^3$$

Dudszus—Danckwardt

$$C_1 = 0.13C_{WP} + 0.87C_{WP}^2 \pm 0.005$$

Murray, for trapezoidal waterlines

$$C_1 = 0.5(3C_{WP} - 1).$$

All the above formulas were successfully implemented in practice; however, they do not directly apply to modern ship hull forms with wetted transom stern, thus some caution is necessary when using them..

2.18.3 Vertical Position of Metacenter

In the early design stages, because the waterplane area coefficient is not known, it is possible to approximate $\overline{KM}$ using other known features of the ship. From the combination of relationships for $\overline{KB}$ and $\overline{BM}$ the following expression is derived:

$$\overline{KM} = B \cdot \left[C_1 \cdot C_2 \cdot \frac{B}{T} \cdot C_M^{-\frac{2}{3}} + \frac{0.9 - 0.3C_M - 0.1C_B}{B/T} \right]$$

where

C_1: describes the waterplane area's lengthwise distribution and its sharpness near the should ers of the ship
$= 0.078$ for waterlines without parallel body
$= 0.083$ for rectangular waterlines (barges)
$= 0.078 + L_P/L_{PP} \cdot 0.005$ generally,

where

L_P: length of parallel body of the waterline, with
$L_P/L_{PP} \cong 0.6 \div 0.7$ for $C_P = 0.8$
$\cong 0.4 \div 0.5$ for $C_P = 0.7$
$\cong 0.2 \div 0.3$ for $C_P = 0.6$
$\cong 0 \div 0.1$ for $C_P = 0.5$

(approximation of L_P when other data are missing)

$$C_2 = \left(C_{WP} / (C_{WP})_{NORM} \right)^a$$

where:

$$(C_{WP})_{NORM} = C_P^{2/3}$$

C_{WP}: given or equal to $(C_{WP})_{NORM}$
$a = 1.5$ for $C_{WP} > (C_{WP})_{NORM}$
$= 1.0$ for $C_{WP} \leq (C_{WP})_{NORM}$

The above standard (norm) waterplane area coefficient $(C_{WP})_{NORM}$ corresponds to nonintense sections of U and V types. However, since there are very pronounced V sections with large flare near the waterline for some ships (e.g., tugboats or modern containerships), it may be assumed that:

$$C_{WP} \cong \left(C_{WP}\right)_{NORM} + 0.05.$$

In addition, for the application of the above formula for $\overline{KM}$, normal hull form and/or stern *without* intense, extended or wetted transom, is assumed.

Finally, it is assumed that the extent of the stern abaft of the aft perpendicular does not exceed 2.5 % of the waterline length (maximum difference between L_{WL} and L_{PP}). If this limit is not observed, the used hull coefficients C_B and C_{WP} should be corrected as follows:

$$C'_B = C_B \cdot L_{PP} / 0.975 L_{WL}$$
$$C'_{WP} = C_{WP} L_{PP} / 0.975 L_{WL}$$

2.18.4 Approximation of Stability at Large Inclination Angles

If during the initial design stage proves necessary to estimate the stability beyond the region of small inclination angles, the restoring arm may be approximated by:

$$h = GZ = (0.5\overline{BM} \cdot \tan^2 \varphi + \overline{GM}) \cdot \sin \varphi$$
$$\approx 0.5\overline{BM} \cdot \varphi^3 + \overline{GM}\varphi \text{ (for small angles } \varphi).$$

The application of the above formula assumes for the hull of the ship:

- vertical sections (wall-sided) around the waterline
- nonimmersion of deck's edge and non-emergence of bottom's bilge extremes.

The formula is valid for angles of $\varphi \leq 10°$ with good accuracy, even for nonvertical sections around the waterline. For larger angles various approximation methods can be used, but their accuracy is not proportional to the required effort for their implementation.

Nevertheless, two useful methods are listed below for further study. They are based on a systematic examination of the influence of the ship's hull form on the ship's stability:

- **Weberling,** Dr.-Ing. thesis, TH Aachen 1974, New Ships 1975.
- **H. E. Guldhammer (**1979**).**

The latter method is based on the well-known, systematic hull form series of FORMDATA, which is widely applied to the hull form design of various types of ships in the last decades.

Table 2.38 Conversion factors of hydrostatic data for geometrically similar ships

Conversion factor $C(\alpha,\beta,\gamma)$	
Waterplane area, A_{WL}	$\alpha\cdot\beta$
Longitudinal position CF of waterline, LCF	A
Longitudinal moment of inertia, I_L	$\alpha^3\cdot\beta$
Transversal moment of inertia, I_T	$\alpha\cdot\beta^3$
Sectional area, A_S	$\beta\cdot\gamma$
Sectional area moment, M_S	$\beta.\gamma^2$
Displacement, ∇	$\alpha\cdot\beta\cdot\gamma$
Longitudinal position CB of buoyancy, $\overline{LCB}$	α
Vertical position CB of buoyancy, $\overline{KB}$	γ
Transversal metacentric radius, $\overline{BM}$	β^2/γ
Longitudinal metacentric radius, $\overline{BM}_L$	α^2/γ
Moment to change trim, MCT	$\alpha^2\cdot\beta$
Force to change displacement TPI (tons per inch change of draught) or TPC (tons per centimeter change of draught)	$\alpha\cdot\beta$
Hull form coefficients, C_B, C_P, C_M, C_{WP}	1

2.18.5 Using the Hydrostatic Data of Similar Ships

Provided that the hydrostatic data of a parent ship are known, for example, the *hydrostatic diagram*s of a ship similar to the one under design, namely, with the same hull form coefficients, similar sectional character, but different main dimensions (*homologous distortion,* see Chap. 4), then the following coefficients can be used to convert the hydrostatic data from the parent ship, subscript 0, to the under design ship, index 1. The method is valid approximately for ships without absolute correspondence in the sectional form, as long as the general character is maintained (Table 2.38).

Longitudinal scale:	$\alpha = L_1/L_0$
Transversal scale:	$\beta = B_1/B_0$
Vertical scale:	$\gamma = T_1/T_0$
General conversion formula:	$(\ )_1 = (\ )_0 \cdot C(\alpha,\beta,\gamma)$.

2.18.6 Effect of Changing the Main Dimensions

During the initial phase of design, the qualitative knowledge of the effect of possible changes of the main particulars on the initial stability, namely, on $\overline{GM}$, is particularly useful.

Assuming that the displacement and the coefficients C_B and C_{WP} do not change, so are the ratios:

$$\overline{KB}/T, \overline{KG}/D_{\kappa\alpha\iota} \overline{BM}/(\overline{BM})_{\mathrm{NORM}}$$
$$\overline{KB}/T, \overline{KG}/D \text{ and } \overline{BM}/(\overline{BM})_{\mathrm{NORM}}$$

where $(\overline{BM})_{\mathrm{NORM}} = B^2/12 \cdot T\, C_B'$, then the following useful expressions according to Munro-Smith (Henschke 1964) are concluded:

$$\frac{\delta(\overline{GM})}{\overline{GM}} = \frac{\overline{KB}}{\overline{GM}} \cdot \frac{\delta T}{T} + 2\left(1 + \frac{\overline{BM}}{\overline{GM}}\right) \cdot \frac{\delta B}{B} - \frac{\overline{BM}}{\overline{GM}} \cdot \frac{\delta T}{T} - \frac{\overline{KG}}{\overline{GM}} \cdot \frac{\delta T}{T}$$

Thus, if we set additionally the draft fixed ($\delta T = 0$), as well as the side depth D ($\delta D = 0$), we obtain:

$$100 \frac{\delta B}{B} = 100 \frac{\delta \overline{GM}}{\overline{GM}} \cdot \frac{1}{2(1 + \overline{BG}/\overline{GM})} [\%]$$

whereas for constant beam ($\delta B = 0$) and $\delta T/T = \delta D/D$ we have:

$$-100 \frac{\delta T}{T} = 100 \frac{\delta \overline{GM}}{\overline{GM}} \cdot \frac{1}{1 + 2\overline{BG}/\overline{GM}} (\%)$$

For the above relationships it has been assumed that the C_B coefficient remains constant. Therefore, as the T or B change, it is assumed that the length changes inversely proportional so as the displacement and C_B to remain fixed.

Now, in case we assume:

$$\nabla, L, C_B, \text{and } C_{WP} \text{ fixed,}$$

and

$$\delta B/B = -\delta T/T = -\delta D/D$$

the following is concluded

$$100 \frac{\delta B}{B} = -100 \frac{\delta T}{T} = 100 \frac{\delta \overline{GM}}{\overline{GM}} \cdot \frac{1}{3 + 4\overline{BG}/\overline{GM}} (\%)$$

In case of

$$\nabla, L, C_B, C_{WP}, \text{ and } D \text{ fixed,}$$

but

$$\delta B / B = -\delta T / T$$

it shows:

$$100\frac{\delta B}{B} = -100\frac{\delta T}{T} = 100\frac{\delta\overline{GM}}{\overline{GM}}\cdot\frac{1}{3+4\overline{BG}/\overline{GM}-\overline{\mathrm{K}G}/\overline{GM}}(\%)$$

Finally, for

$$\nabla, L, T, D, \text{ and } C_{WP} \text{ fixed}$$

as well as

$$\delta B / B = -\delta C_{\mathrm{B}} / C_{\mathrm{B}}$$

the following is concluded:

$$100\frac{\delta B}{B} = 100\cdot\delta\overline{KM} / \left(\frac{\nabla}{3A_{\mathrm{WL}}} + 3\overline{BM}\right)(\%)$$

2.18.7 Typical Values of Metacentric Height

In the context of verification/examination of the *initial stability* during the conceptual/preliminary design stage of a ship, it is usually sufficient to compare the resultant *GM* value with some typical values of similar types of ships, as shown in Table 2.39.

High $\overline{GM}$ values ensure satisfactory stability and safety for the ship against capsize only if they are accompanied by a sufficient range of positive restoring arm curve for large inclination angles; it should be noted, however, that large $\overline{GM}$

Table 2.39 Typical $\overline{GM}$ values for modern ships in the departure, full load condition

General cargo ships	>0.4–0.9 m
Containerships	>0.3–0.6 m
Short-sea cargo ships	>0.4–1.0 m
Tankers	1.0–6.0 m
Bulk carriers	0.6–2.0 m
Reefer ships	0.7–1.1 m
Tug boats	0.8–1.3 m
Fishing vessels	0.7–1.2 m
Passenger ships (oceangoing)	1.0–2.5 m
Passenger ships (limited waters)	0.5–1.5 m
Passenger CATAMARAN ships	>10 m
SWATH type Passenger ships	1.5–2.5 m

Fig. 2.96 Shift of deck containers due to excessive transverse accelerations

values trigger intense *roll motions* and *transverse accelerations* on the ship's deck (and higher positions), in view of the relationship:

$$T_{\text{Roll}} \propto B/\sqrt{\overline{GM}}$$

where T_{Roll}: natural roll period of the ship.

For large values of $\overline{GM}$, that is, small roll period T_{Φ}, the resultant transverse acceleration[33] on the ship's deck (and higher positions) in *resonance* situation (i.e., for wave excitation period close to the ship's natural roll period), becomes particularly pronounced resulting in nausea or injuries of passengers and crew, the shift or damage of higher up stacked cargo (e.g., deck containers, Fig. 2.96, shift of vehicles onboard Ro-Ro ships, etc).

In conclusion, it is recommended that the $\overline{GM}$ values should not be unreasonably high, but certainly, in any case, regardless of the type and size of the ship, not to be less than about 0.30–0.35 m in departure and design loading condition.

[33] The transverse acceleration at certain position of a rolling ship is proportional to the distance of the reference point from the ship's rolling axis (which is assumed passing near the ship's mass center), and is inversely proportional to the *square* of T_{Φ} (or directly proportional to square of the circular roll frequency $\omega_{\Phi}=2\pi/T_{\Phi}$). Obviously, the transverse acceleration increases with larger distances from ship's roll axis and lower values of T_{Φ}.

2.18.8 Verification of Stability

The verification of a satisfactory status ship's stability refers to the sufficiency of the ship's stability (and floatability) in intact and damage condition with respect to the requirements of specified stability criteria, as laid down in regulations developed and approved by the International Maritime Organisation (IMO). We will be limiting in the following our deliberations to the intact stability criteria, as necessary in the frame of the ship's preliminary design, and refer to Appendix E with respect to the evolution of the criteria for the ship in damage condition.

IMO's Maritime Safety Committee adopted in its 85th session the presently valid International Code on Intact Stability, 2008 (2008 Intact Stability Code, IMO 2008c), taking into account technical developments to update the 1993 Intact Stability Code (resolution A.749(18)) and later amendments thereto (resolution MSC.75(69)). The 2008 IS Code provides, in a single document, both mandatory requirements and recommended operational provisions relating to intact stability, like general precautions against capsizing (criteria regarding metacentric height (GM) and righting lever (GZ)); weather criterion (severe wind and rolling criterion); effect of free surfaces and icing; and watertight integrity.

The IS2008 Code contains intact stability criteria for the following types of ships and other marine vehicles of 24 m in length and above, unless otherwise stated:

1. cargo ships;
2. cargo ships carrying timber deck cargoes;
3. passenger ships;
4. fishing vessels;
5. special purpose ships;
6. offshore supply vessels;
7. mobile offshore drilling units;
8. pontoons; and
9. cargo ships carrying containers on deck and containerships.

The below *general criteria* regarding the properties of the righting arm curve in intact condition apply to all ships, except for stated otherwise:

a. The area under the righting lever curve (GZ curve) shall not be less than 0.055 m-radians up to $\varphi = 30°$ angle of heel and not less than 0.09 m-radians up to $\varphi = 40°$ or the angle of down-flooding φ_f[34], if this angle is less than 40°.
b. Additionally, the area under the righting lever curve (GZ curve) between the angles of heel of 30° and 40° or between 30° and φ_f, if this angle is less than 40°, shall not be less than 0.03 m-rad.
c. The righting lever GZ shall be at least 0.2 m at an angle of heel equal to or greater than 30°.

[34] φ_f is an angle of heel at which openings in the hull, superstructures or deckhouses which cannot be closed weathertight immerse. In applying this criterion, small openings through which progressive flooding cannot take place need not be considered as open.

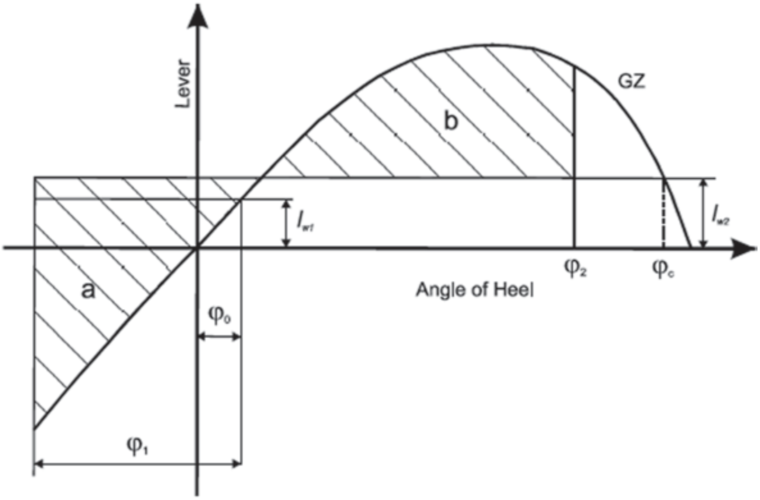


Fig. 2.97 Severe wind and rolling (weather criterion)

d. The maximum righting lever shall occur at an angle of heel not less than 25°. If this is not practicable, alternative criteria, based on an equivalent level of safety[35], may be applied subject to the approval of the administration.
e. The initial metacentric height GM_0 shall not be less than 0.15 m.

The below *weather criterion* considers the case of severe wind and excessive rolling motions due to the excitation of incoming waves (refer Fig. 2.97) and applies to *special ship types and to floating vehicles*:

10. The ship is subjected to a steady wind pressure acting perpendicular to the ship's centerline which results in a steady wind heeling lever (l_{w1});
11. from the resultant angle of equilibrium (φ_0), the ship is assumed to roll due to wave action to an angle of roll (φ_1) to windward. The angle of heel under action of steady wind (φ_0) should not exceed 16° or 80 % of the angle of deck edge immersion, whichever is less;
12. the ship is then subjected to a gust wind pressure which results in a gust wind heeling lever (l_{w2}); and
13. under these circumstances, area *b* shall be equal to or greater than area *a,* as indicated in Fig. 2.97 below:

where the angles in Fig. 2.97 are defined as follows:

φ_0 = angle of heel under action of steady wind
φ_1 = angle of roll to windward due to wave action
φ_2 = angle of down-flooding (φ_f) or 50° or φ_c, whichever is less,

[35] Refer to the Explanatory Notes to the International Code on Intact Stability, IMO 2008a (MSC.1/Circ.1281)

where:

φf = angle of heel at which openings in the hull, superstructures or deckhouses which cannot be closed weathertight immerse. In applying this criterion, small openings through which progressive flooding cannot take place need not be considered as open

φc = angle of second intercept between wind heeling lever l_{w2} and GZ curves.

The wind heeling levers l_{w1} and l_{w2} referred to in points 1 and 3 above are constant values at all angles of inclination and shall be calculated as follows:

$$l_{\mathrm{w1}} = \frac{P * A * Z}{1{,}000 * g * \Delta}(\mathrm{m})$$

$$l_{\mathrm{w2}} = 1.5 * l_{\mathrm{w1}}(\mathrm{m})$$

where:

P = wind pressure of 504 Pa. The value of P used for ships in restricted service may be reduced subject to the approval of the administration

A = projected lateral area of the portion of the ship and deck cargo above the waterline (square meter)

Z = vertical distance from the center of A to the center of the underwater lateral area or approximately to a point at one half the mean draught (meter)

Δ = displacement (tons)

g = gravitational acceleration of 9.81 m/s^2.

Alternative means for determining the wind heeling lever (l_{w1}) may be accepted, to the satisfaction of the administration.

The angle of roll (φ_1) referred to in point 2 above shall be calculated as follows:

$$\varphi_1 = 109 * k * X_1 * X_2 * \sqrt{r * s}\,(\text{degrees})$$

where:

X_1 = factor as shown in Table 2.40

X_2 = factor as shown in Table 2.40

k = factor as follows:

k = 1.0 for round-bilged ship having no bilge or bar keels

k = 0.7 for a ship having sharp bilges

k = as in Table 2.40 for a ship having bilge keels, a bar keel or both

$$r = 0.73 + 0.6(OG/d)$$

Table 2.40 Values of factor X_1, X_2, k, and s

B/d	X_1	C_B	X_2	$\frac{A_k * 100}{L_{WL} * B}$	k	T	s
≤2.4	1.0	≤0.45	0.75	0	1.0	≤6	0.100
2.5	0.98	0.50	0.82	1.0	0.98	7	0.098
2.6	0.96	0.55	0.89	1.5	0.95	8	0.093
2.7	0.95	0.60	0.95	2.0	0.88	12	0.065
2.8	0.93	0.65	0.97	2.5	0.79	14	0.053
2.9	0.91	≥0.70	1.00	3.0	0.74	16	0.044
3.0	0.90			3.5	0.72	18	0.038
3.1	0.88			≥4.0	0.70	≥20	0.035
3.2	0.86						
3.4	0.82						
≥3.5	0.80						

Intermediate values in these tables shall be obtained by linear interpolation

with:

$OG = KG - d,$

d = mean moulded draught of the ship (meter)
S = factor as shown in Table 2.40, where T is the ship roll natural period.

In absence of sufficient information, the following approximate formula can be used:

Rolling period

$$T = \frac{2 * C * B}{\sqrt{GM}} \cdot (\text{second})$$

where:

$$C = 0.373 + 0.023(B/d) - 0.043(L_{wl}/100).$$

The symbols in Table 2.40 and the formula for the rolling period are defined as follows:

L_{wl} = length of the ship at waterline (meter)
B = moulded breadth of the ship (meter)
d = mean moulded draught of the ship (meter)
C_B = block coefficient (−)
A_k = total overall area of bilge keels, or area of the lateral projection of the bar keel, or sum of these areas (square meter)
GM = metacentric height corrected for free surface effect (meter).

Special Criteria for Certain Types of Ships

Passenger Ships

Passenger ships shall comply with the *general criteria and the weather criterion requirements*. In addition, the angle of heel accounting for the crowding of passengers to one side as defined below, shall not exceed 10°.

A minimum weight of 75 kg shall be assumed for each passenger except that this value may be increased subject to the approval of the administration. In addition, the mass and distribution of the luggage shall be approved by the administration. The height of the center of gravity for passengers shall be assumed equal to:

- 1 m above deck level for passengers standing upright; account may be taken, if necessary, of camber and sheer of deck
- 0.3 m above the seat in respect of seated passengers.

In addition, the angle of heel account for turning maneuver shall not exceed 10° when calculated using the following formula:

$$M_{\mathrm{R}} = 0.200 * \frac{v_0^2}{L_{\mathrm{WL}}} * \Delta * \left(KG - \frac{d}{2} \right)$$

where:

M_{R} = heeling moment (kilonewton-meter)
V_{o} = service speed (meter per second)
L_{WL} = length of ship at waterline (meter)
Δ = displacement (tons)
D = mean draught (meter)
KG = height of center of gravity above baseline (meter).

Oil tankers of 5,000 t DWT and above

Oil tankers[36] shall comply with the provisions of regulation 27 of Annex I to MARPOL 73/78 (which lead to the same *general intact stability requirements* on GZ, as outlined above) .

Cargo ships carrying timber deck cargoes

Cargo ships carrying timber deck cargoes shall comply with the *general criteria and the weather criterion requirements* unless the administration is satisfied with the application of alternative provision, laid down in the IS2008 code.

Cargo ships carrying grain in bulk

The intact stability of ships engaged in the carriage of grain shall comply with the requirements of the International Code for the Safe Carriage of Grain in Bulk adopted by resolution MSC.23(59).

[36] Oil tanker means a ship constructed or adapted primarily to carry oil in bulk in its cargo spaces and includes combination carriers and any chemical tanker as defined in Annex II of the MARPOL Convention when it is carrying a cargo or part cargo of oil in bulk.

High-speed craft

High-speed craft[37] constructed on or after 1 January 1996 but before 1 July 2002, to which chapter X of the 1974 SOLAS Convention applies, shall comply with stability requirements of the 1994 HSC Code (resolution MSC.36(63)). Any high-speed craft to which chapter X of the 1974 SOLAS Convention applies, irrespective of its date of construction, which has undergone repairs, alterations or modifications of a major character; and a high-speed craft constructed on or after 1 July 2002, shall comply with stability requirements of the 2000 HSC Code (resolution MSC.97(73)).

Containerships greater than 100 m

Requirements for containerships[38] over 100 m in length regarding the GZ curve properties are as following:

a. The area under the GZ curve should not be less than $0.009/C$ m rad up to $\varphi = 30°$ angle of heel, and not less than $0.016/C$ m rad up to $\varphi = 40°$ or the earlier defined angle of flooding φ_f, if this angle is less than $40°$.
b. Additionally, the area under the GZ curve between the angles of heel of 30° and 40° or between 30° and φ_f, if this angle is less than 40°, should not be less than $0.006/C$ m rad.
c. The righting lever GZ should be at least $0.033/C$ m at an angle of heel equal or greater than 30°.
d. The maximum righting lever GZ should be at least $0.042/C$ m.
e. The total area under the righting lever curve (GZ curve) up to the angle of flooding φ_f should not be less than $0.029/C$ m rad.

Since the criteria in this section were empirically developed with the data of containerships less than 200 m in length, they should be applied to *ships beyond such limits with special care*. In the above criteria the form factor C should be calculated using the below formula and the definitions of Fig. 2.98:

$$C = \frac{dD'}{B_m^2}\sqrt{\frac{d}{KG}}\left(\frac{C_B}{C_W}\right)^2\sqrt{\frac{100}{L}}$$

where:

d = mean draught (meter)
D' = moulded depth of the ship, corrected for defined parts of volumes within the hatch coamings according to the formula:

$$D' = D + h\left(\frac{2b - B_D}{B_D}\right)\left(\frac{2\Sigma l_H}{L}\right)$$

[37] *High-speed craft* (HSC) is a craft capable of a maximum speed, in meter per second (m/s), equal to or exceeding: $3.7 * \nabla^{0.1667}$, where: ∇ = displacement volume corresponding to the design waterline (cubic meter).

[38] They may also be applied to other cargo ships in this length range with considerable flare or large water plane areas.

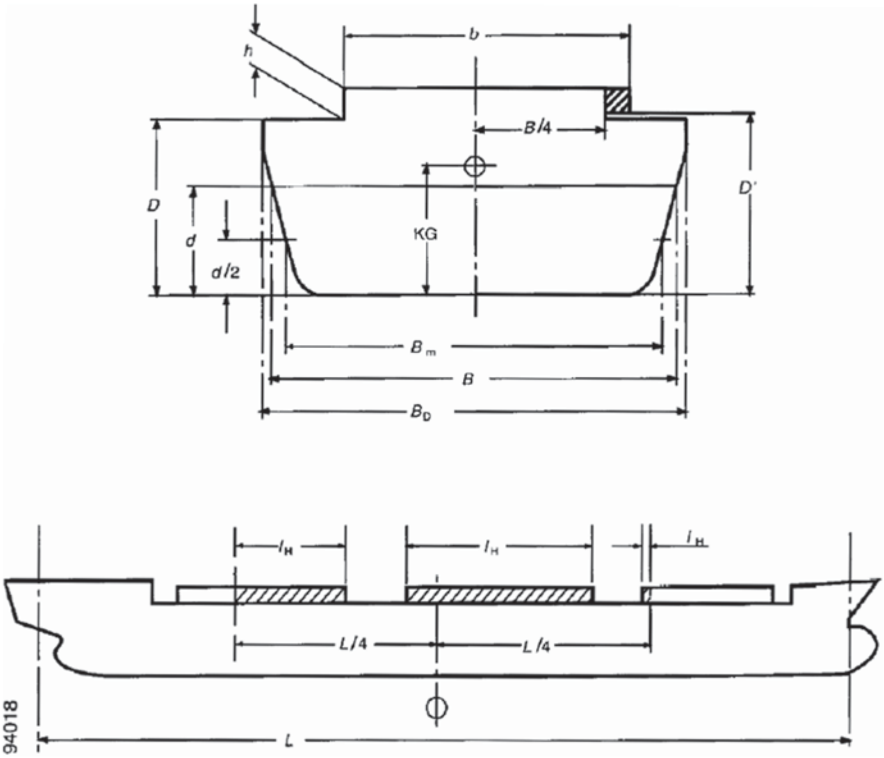


Fig. 2.98 Definition of parameters for the intact stability form factor C of containerships. (The *shaded* areas in Fig. 2.98 represent partial volumes within the hatch coamings considered contributing to resistance against capsizing at large heeling angles when the ship is on a wave crest. The use of electronic loading and stability instrument is encouraged in determining the ship's trim and stability during different operational conditions)

D = moulded depth of the ship (meter);
B_D = moulded breadth of the ship (meter);
KG = height of the center of mass above base, corrected for free surface effect, not be taken as less than d (meter);
C_B = block coefficient;
C_W = water plane coefficient;
l_H = length of each hatch coaming within $L/4$ forward and aft from amidships (meter);
b = mean width of hatch coamings within $L/4$ forward and aft from amidships (meter);
h = mean height of hatch coamings within $L/4$ forward and aft from amidships (meter);
L = length of the ship (meter);
B = breadth of the ship on the waterline (meter);
B_m = breadth of the ship on the waterline at half mean draught (meter).

2.18.9 Verification of Trim and Bow Height

We assume that in the design loading condition the ship will not present any undesirable trim[39], may be by taking a limited amount of ballast water[40].

From the well-known relationship for the trim at small angles:

$$t = L \cdot \frac{M_t}{\Delta \cdot \overline{GM}_L}$$

where M_t: trim moment, $\overline{GM}_L$: longitudinal metacentric height,

it is concluded with

$$\overline{GM}_L \cong \overline{BM}_L + T/2$$

and

$$\overline{BM}_L \cong 0.07 \cdot L^2 / T \,(\text{Schneekluth})$$

where

$$L \equiv L_{WL}$$

the resulting trim t for nonbalanced trim moments M_t.

Regardless of the existence of trim, the minimum height of the bow of the ship, as the one defined at the ship's forward perpendicular (see Fig. 2.99), is specified in the International Load Line Convention:

Minimum bow height

Ships with $L<250$ m:

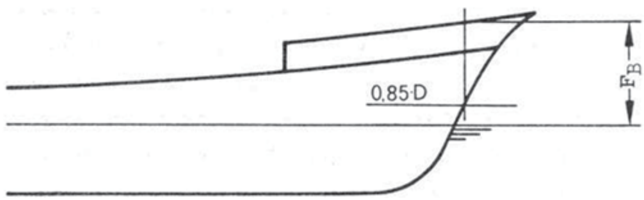


Fig. 2.99 Bow height F_B at the *forward perpendicular*

[39] A small stern (rarely bow down) trim is often desirable and generally acceptable.

[40] A significant amount of ballast water in the design loading condition may be necessary for some types of ships, like containerships, carrying a significant number of containers on deck. Modern ship design concepts aim at significantly reducing the amount of ballast water both in the design load and the ballast condition, thus reducing both fuel cost and incurring additional cost for ballast water treatment in view of IMO's guidelines on ballast water management (latest, IMO-MEPC.173(58), IMO 2008b).

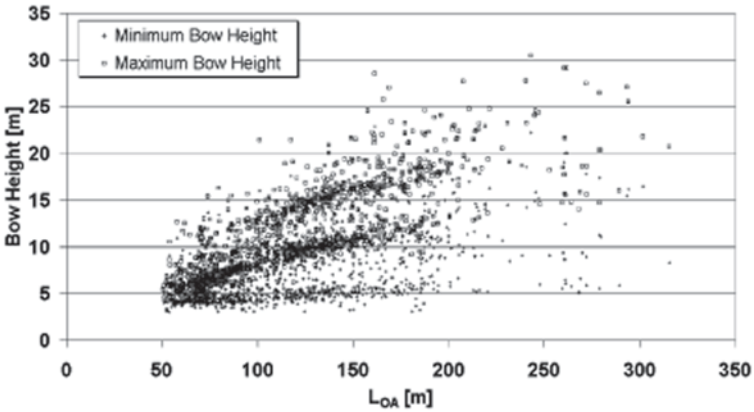


Fig. 2.100 Statistics of bow heights of passenger/ferries with forecastle. (Tagg et al. 2001)

$$F_B \geq 56 \cdot L\left(1-\frac{L}{500}\right) \cdot \frac{1.36}{C_B+0.68}$$

where F_B (millimeter), L (meter).

Ships with $L \geq 250$ m:

$$F_B \geq 7000 \cdot \frac{1.36}{C_B+0.68}$$

In both formulas, the minimum C_B is considered as equal to 0.68.

The aforementioned formulae apply to existing ships in accordance with the old Load Line regulations (see Sect. 2.19 for the most recent changes). In the above Fig. 2.100, we can observe statistical values of bow heights of passenger/ferry ships according to Tagg et al. (2001):

Concluding remarks on the verification of stability and trim

It is clear from the above deliberations in this section that the verification/examination of the ship's stability and trim during the preliminary design stage is limited to the control of the ship's behavior in intact condition and for small inclination angles (*initial stability and trim*). The examined values are determined by the wetted (buoyant) part of the ship's hull (in calm water). In order to examine the stability of the ship at large inclination angles, the knowledge of the ship's hull above the design waterline[41], including her freeboard, are necessary. Finally, for examining the ship's damage stability, the internal subdivision of the ship, including the position of watertight transverse and longitudinal bulkheads, of decks, openings and down-flooding points, is required.

[41] Including the location of nonwatertight openings of the ship's outer shell.

2.19 Freeboard and Sheer

2.19.1 Factors Affecting the Freeboard

- Large freeboard ensures large reserve buoyancy and increased the ship survivability in case of hull damage. It also improves the ship stability at large inclination angles.
- Sufficient freeboard improves the ship's behavior in seaways. Particularly, it provides improved safety against wetting of the deck, damage of deck cargo (deck containers) and likely water ingress into the ship's holds from incoming, high waves.
- Because of the pitch-axis location in general abaft of the midship section and consequently the more intense bow motions, a higher freeboard at forward perpendicular is required.
- This increased freeboard is also necessary in the "critical region" around the ship's forward perpendicular, namely, for approximately 15 % of the ship's length, which is so specified in the Load Line Convention[42].

The bow height, measured at forward perpendicular between design waterline (summer load line) and the ship's weathertight deck (e.g., forecastle), rarely exceed 8–9 % of the length L_{pp}. Generally, this percentage reduces with the increase of the absolute ship size. Fast ships need to have relatively higher bows, compared to the slow ones, because of higher "swell-up" of the generated bow wave and likely more intense bow motions.

The following figure presents the statistics of freeboard heights for various shiptypes on the basis of data of of IHS Fairplay World Shipping Encyclopedia, v. 12.01, 2011 (Fig. 2.101).

The latest Load Line Regulations (ICLL 1988, Regulation 39) specify *as minimum freeboard* at the forward perpendicular, for normal trim, the following:

$$F_\mathrm{b} = \left(6075\left(\frac{L}{100}\right) - 1875\left(\frac{L}{100}\right)^2 + 200\left(\frac{L}{100}\right)^3\right)$$
$$\times\left(2.08 + 0.609C_\mathrm{b} - 1.603C_\mathrm{wf} - 0.0129\left(\frac{L}{d_1}\right)\right)$$

[42] The International Load Line Convention has a long history, starting in 1890, when the first rules for a minimum freeboard for all ships departing from *British ports* (thanks to the British politician *Samuel Plimsoll*) were established. The first form of relevant *international regulations* was agreed in 1930 by 54 countries. In the framework of the International Maritime Organization (IMO), the first International Convention on Load Lines (ICLL) was first approved on 5 April 1966 and entered into force on 21 July 1968. Some changes followed in 1971, 1975, 1979, 1983, and 1995, which never entered into force because of lack of enough flag state acceptances; the 1966 ICLL provisions were amended by the adopted Protocol of 1988, which entered into force on 3 February 2000. The intention of the Protocol of 1988 was to harmonize the requirements of the Convention on the survey and certification with the corresponding requirements of SOLAS & MARPOL 73/78. The Protocol of 1988 was once more amended by the 2003 Amendments, which were adopted with the Resolution MSC.143 on 5 June 2003 and entered into force on 1 January 2005, as well as with further Amendments adopted with the Resolution MSC.172 on 9 December 2004 and which entered into force on 1 July 2006.

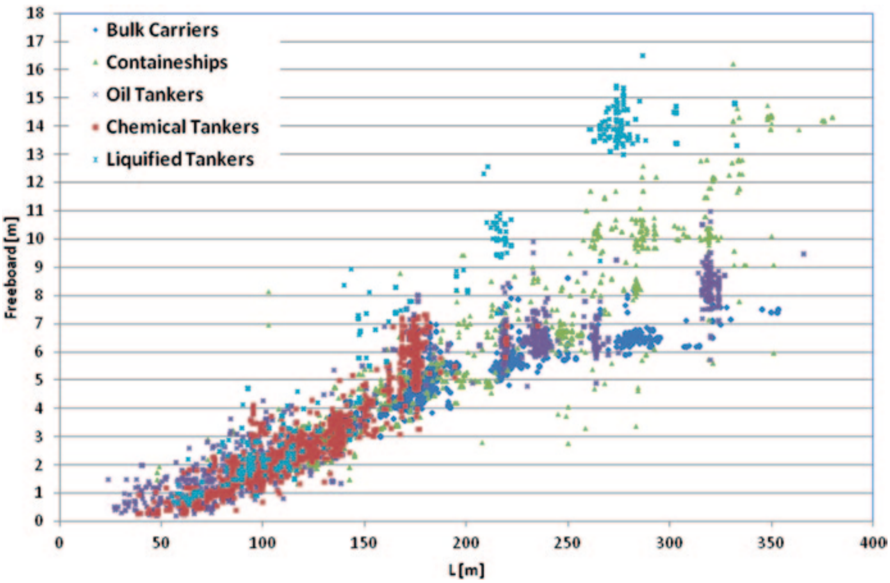


Fig. 2.101 Statistics of freeboard height of dry cargo and liquid cargo ships (analysis of data of IHS Fairplay 2011)

where

F_b (millimeter):	minimum bow height
L (meter):	length for freeboard calculation
B (meter):	breadth for freeboard calculation
d_1 (meter):	draft at 85 % of the side depth D
C_b:	block coefficient according to Regulation 3
C_{wf}:	waterplane area coefficient of the fore body (from midship to forward).

$$C_{wf} = \frac{A_{wf}}{\frac{L}{2} \cdot B}$$

A_{wf} (square meter): waterplane area of the fore body at draft d_1.

The ICLL regulations state that if the minimum value of the bow height at FP is achieved by consideration of a sheer, then the same height must extend over at least 15 % L from the forward perpendicular. In addition, if the height is measured with respect to an existing forecastle, then it is appropriate for such a forecastle to extend over at least 7 % L aft of FP.

Similar specifications for a minimum height of the ship's stern do not exist. However, it is assumed that the resulting height will be at least equal to the freeboard at the ship's midship section. Furthermore, if the above bow height is achieved as freeboard at the ship's midship section, it is obvious that generally the provision of an additional sheer at the freeboard deck for the satisfaction of Load Line Regulations is not required.

Table 2.41 Typical values of bow height and of height of the strength deck for various types of merchant ships; synthesis of data by E. Strohbusch (1971) and partly revised according to IHS Fairplay World Shipping Encyclopedia, v. 12.01, 2011)

Ship type	L_{PP}/D	F_{FP} (m)	F_{FP}–%L_{PP}
Fast seagoing cargo ships	9.9–13.5	13.0–18.5	4.9–7.5
Slow seagoing cargo ships		12.0–13.0	6.3–7.9
Coastal cargo ships	10.0–12.0	3.5–4.5	Up to 7.0
Small short sea passenger ships	10.4–11.6	6.0–7.0	6.6–7.9
Ferries	8.6–10.3	7.0–10.0	7.0–10.0
Fishing vessels	8.2–9.0	5.0–6.5	8.0–8.5
Tugboats	7.7–10.0	4.6–7.4	8.2–12.0
Bulk carriers	10.5–12.8	4.4–4.9	8.8–10.5
Tankers	12.0–14.0	3.6–4.5	9.4–11.7
Fast seagoing reefers	~11.0	5.6–6.6	7.2–8.8

Typical values for the bow height and the height of the strength deck (which is not necessarily the freeboard deck) for common types of merchant ships are listed in Table 2.41.

2.19.2 Verification of Freeboard

The calculation and verification of the allowable freeboard, namely of the permitted vertical distance of the upper edge of the freeboard deck (typically: uppermost continuous and watertight deck) from the upper edge of the corresponding load line (generally: at the design draft of the ship), are governed by the regulations of the International Convention on Load Lines and *determine the maximum allowable loading draft of the ship. Naval ships, fishing vessels and boats of length smaller than 24 m* are *generally exempted* from the implementation of these regulations. A numerical example of the application of the ICLL regulations to cargo ships is given in reference Papanikolaou (2009a, Vol. 2).

In the initial design stage, the examination of the freeboard aims at verifying the compatibility of the initially estimated principal dimensions and of other fundamental ship values, such as of the ship's length L, side depth *D*, draft *T,* block coefficient *C*B, and of the extent/type of the ship's superstructures. In particular, the validity of the selection of the ship's side depth *D* is confirmed by the simultaneous control of the following ship characteristics:

- hold volume (see Sect. 2.17)
- freeboard (see current paragraph, Sect. 2.19.2)
- ICLL regulations
- stability (see Sect. 2.18).

The ultimate objective is to achieve a minimum, buy sufficient freeboard and to ensure satisfactory reserve buoyancy in case of hull damage and internal flooding. The corresponding freeboard deck, from which the freeboard is measured, is in general identical to upper watertight boundary of ship's watertight bulkheads (*bulkhead*

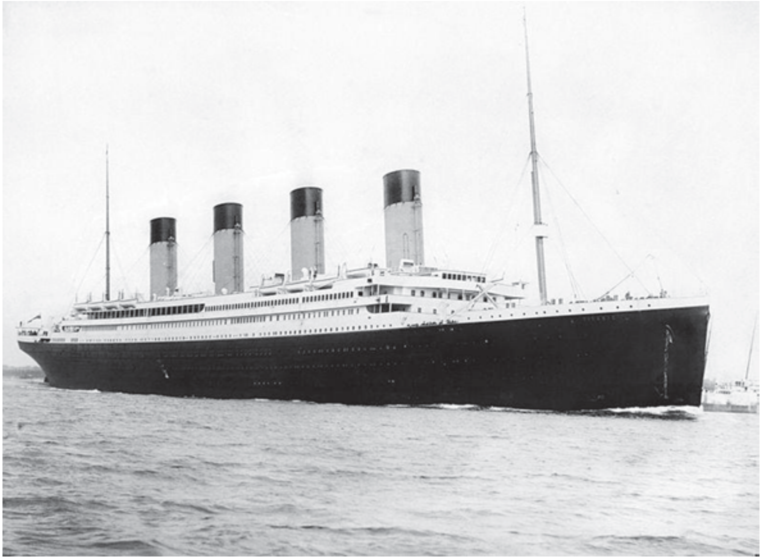


Fig. 2.102 RMS *Titanic* departing Southampton on 10 April 1912 (last voyage)

deck)[43]. For tankers (category "A" ships according to ICLL, namely, ships carrying exclusively liquid cargo), *reduced freeboards* are specified according to the regulations, because of the small permeability of the fully loaded ships (in case of hull breach) and of their watertight subdivision. Reduced freeboards are also specified exceptionally for bulk carriers, if proven to be safe (do not sink or capsize) in the case of flooding of *one (B-60 ships)* or *two neighboring (B-100 ships)* compartments, except for the engine room.

It is noted that for Ro-Ro passenger ships the freeboard deck is identical to the ship's main car deck (which is also the ship's bulkhead deck) (Fig. 2.102)[44].

Satisfactory freeboard allows:

- Prevention of deck wetness and entrance of water into the ship through unprotected or nonwatertight openings

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[43] The ICLL regulations define the *freeboard deck* as the *uppermost continuous deck of the ship*, which is exposed to the weather and the sea. Thus, the freeboard deck is at *least weathertight,* but generally also *watertight*. Exceptionally, the authorities may permit the freeboard deck to be a lower deck (and not the uppermost, continuous deck), which must be continuous between the peak ballast tanks of the ship (fore and aft-peak bulkheads). In this case the space above this lower placed freeboard deck and up to the deck above it is treated *as superstructure*.

[44] [synthesis from Wikipedia] RMS *Titanic* was a British passenger liner that sank in the North Atlantic Ocean on 15 April 1912 after colliding with an iceberg during her maiden voyage from Southampton, UK to New York City, USA. The sinking of *Titanic* caused the deaths of 1,502 people in one of the deadliest peacetime maritime disasters in modern history. On her maiden voyage, she carried 2,224 passengers and crew. The RMS *Titanic* was the largest ship afloat at the time of her maiden voyage and was thought to be unsinkable due to her very dense subdivision. *She was lacking, however, a watertight bulkhead-deck and this was the main reason for her sinking*. One of their most important legacies was the establishment in 1914 of the International Convention for the Safety of Life at Sea (SOLAS), which still governs maritime safety today.

- Protection of crew working on deck
- Safety of cargo stowed on deck (e.g., deck containers)
- Increase of the range of stability for large inclination angles
- Satisfactory stability in damage condition.

If during the control of the ship's freeboard a significant failure is identified, which cannot be compensated with small design corrections, such as changing of deck sheer, small changes in the extent of superstructures, it is always recommended to increase the side depth *D*. However, as the steel weight will simultaneously slightly increase, this will result to a larger draft, compared to the original one, so that the change of *D* cannot be fully transferred to a "gain" in terms of freeboard. On the other side, if during the examination of the hold volume the space proves sufficient, then the proposed increase of *D* can be accompanied by a corresponding reduction of length *L*; the latter will eventually result in a reduction of the required freeboard (see Table 2.42 of basic freeboards), while the structural weight of the ship may

Table 2.42 Freeboard table according to ICLL

Length of ship (m)	Freeboard for type "A" ships (mm)	Freeboard for type "B" ships (mm)
24	200	200
30	250	250
40	334	334
50	443	443
60	573	573
70	706	721
80	841	887
90	984	1,075
100	1,135	1,271
110	1,293	1,479
120	1,459	1,690
130	1,632	1,901
140	1,803	2,109
150	1,968	2,315
160	2,126	2,520
170	2,268	2,716
180	2,393	2,915
190	2,508	3,098
200	2,612	3,264
210	2,705	3,430
220	2,792	3,586
230	2,875	3,735
240	2,946	3,880
250	3,012	4,018
260	3,072	4,152
270	3,128	4,276
280	3,176	4,397
290	3,220	4,513
300	3,262	4,630
310	3,298	4,736
320	3,331	4,844
330	3,358	4,955
340	3,382	5,055
350	3,406	5,160
360	3,425	5,260
365	3,433	5,303

Freeboards at intermediate lengths of ship shall be obtained by linear interpolation

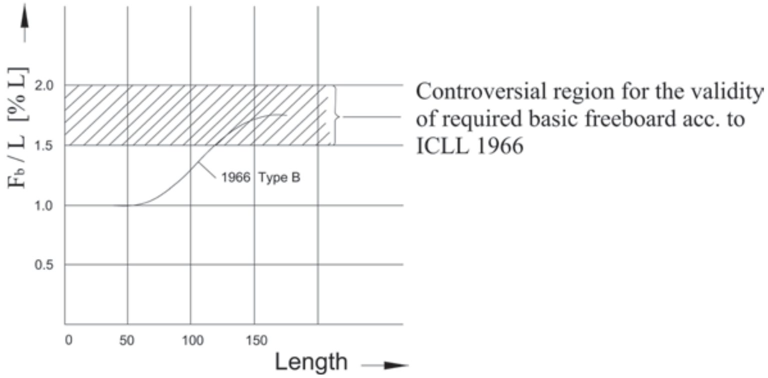


Fig. 2.103 Basic freeboard for ships of category B.

remain fixed or be even reduced. It is assumed that the increase of D and the consequent increase of the center of mass of the ship $\overline{KG}$, do not create significant initial stability problems ($\overline{GM}$ requirement) for the study ship (see Sect. 2.18).

In general it can be concluded that *"volume carriers"* (see Sect. 1.3.7.2) due to the nature of the transferred cargo (low specific weight, high stowage factor) rarely exhibit problems on satisfying the requirements of the Load Line Regulations, namely, they do not fully exploit the allowable margin of draft, in terms of the Load Line requirements (or their actual freeboard is larger than the minimum one). On the contrary, "*deadweight carriers*,", which carry relatively heavy cargoes and for which the adequacy of hold volume is not an issue (e.g., tankers and bulkcarriers), reach the limit of *minimum allowable freeboard* of the Load Line Convention in terms of their design draft. The same is often valid for Ro-Ro passenger ships carrying heavy trucks, especially for those which are conversions of originally other types of ships.

Existing regulations appear to penalize the relatively large ships (or favor smaller ships) since the specified values for the *basic freeboard* for small ships ($L \leq 65$ m) is less than/equal to 1 % L, while the corresponding required height for ships with approximate $L \geq 120$ m is more than 1.5 % L (see Fig. 2.103 and critical review, Sect. 2.19.4).

Simplified calculation of freeboard

In the context of conceptual/preliminary ship design, the accurate calculation of the required freeboard in accordance with the ICLL regulations presents difficulties due to the unavailability of certain necessary data.

If comparable data from similar ships are used to determine the ship's principal dimensions, it is rational to assume that the estimated ratio (D/T) will be a guide for the determination of the anticipated freeboard, without of course excluding differentiations with respect to the implementation of precise regulations to that ship.

Table 2.43 Corrections a_1, for the simplified calculation of the freeboard of general cargo ships without superstructure amidships by Danckwardt

	a_1	
L_{PP} (m)	$C_B \leq 0.68$	$C_B = 0.80$
150	1.335	1.36
140	1.335	1.34
130	1.298	1.32
120	1.280	1.30
110	1.261	1.28
100	1.243	1.26
90	1.225	1.24
80	1.206	1.22
70	1.188	1.20
60	1.170	1.18

For the initial approximation of the freeboard of general cargo ships, the following simplified method by Danckwardt (Henschke 1964) can be applied[45]. In this method the ratio (D/T) is calculated as follows:

1. Ships with forecastle but *without* superstructure amidships:

$$D/T = a_1 - 0.10\left(l_S / L_{PP}\right)$$

where

a_1 $= f(L_{PP}, C_B)$, see Table 2.43
l_S = overall length superstructures between the perpendiculars

Remarks:

i. For C_B values between 0.68 and 0.80 it is proposed to interpolate the values in the table.
ii. For lengths L_{PP} and coefficients C_B significantly beyond the given limits ($L_{PP} = 150$ m and $C_B = 0.80$) *extrapolation* is *not* recommended.

2. Ships with forecastle *and* superstructure amidships:

where

a_2 $= f(L_P L/D,\ C_B)$, see Table 2.44,
b_2 $= f(L/D,\ C_B)$, see Table 2.44.

The above method can be best used for general cargo ships and relatively small tankers/bulkcarriers with good results (according to Danckwardt ±2%).

[45] The method refers actually to the "three island" ship concept, characteristic to ships with forecastle, bridge/superstructure amidships and stern poop.

Table 2.44 Corrections a_2 and coefficients b_2 for the simplified calculation of the freeboard of general cargo ships with forecastle and superstructure amidships by Danckwardt

L_{pp} [m]	C_B 0.68			C_B=0.80		
	L/D			L/D		
	10	12.5	15	10	12.5	15
150	1.370	1.350		1.396	1.408	1.428
140	1.357	1.337		1.381	1.389	1.402
130	1.342	1.322		1.364	1.369	1.376
120	1.324	1.310		1.343	1.350	1.350
110	1.298	1.293		1.320	1.327	1.325
100	1.271	1.269		1.294	1.302	1.300
90	1.245	1.242		1.262	1.272	1.276

Corrections b_2

L/D	C_B 0.68	C_B=0.80
10	0.152	0.150
12.5	0.170	0.200
15	0.202	0.224

Alternatively, it is suggested to use the following simplified diagrams that take into account only the main corrections on the basic freeboard resulting from the Regulations (see Figs. 2.104, 2.105, 2.106, 2.107, 2.108, 2.109, 2.110, and 2.111 according to Danckwardt). The freeboard of dry cargo and liquid cargo ships (tankers) is given as a function of length L_{PP}, the L/D ratio and a presumed normal extent of the superstructure (l_S/L_{PP}). Because the diagrams are for standard block coefficient $C_{B(0.85D)}=0.68$, the values need to be increased according to the corrections of Figs. 2.110 and 2.111.

2.19.3 Sheer

- *Application criteria*: The existence of a sheer on the upper decks of the ship, that is, an upward slope of the centerline of the ship's deck from amidships towards the ends, significantly improves the seakeeping characteristics of the ship and increases the reserve buoyancy at the ends. In view of this, earlier built ships were all designed with sheer. Newer buildings, particularly tankers, bulkcarriers, containerships, Ro-Ro, etc., do not dispose a sheer, thus simplifying the construction (reducing building cost) or for operational reasons (e.g., car ferries: problems with the lashing/fastening of vehicles). However, it is possible to have straight line sheer (instead of the parabolic type) at the ends of the ship, for example, in the forecastle region, what still improves the ship's seakeeping behavior in waves, whereas the ship's construction remains in this respect simple. Particularly for small ships with special requirements on seaworthiness, such as fishing

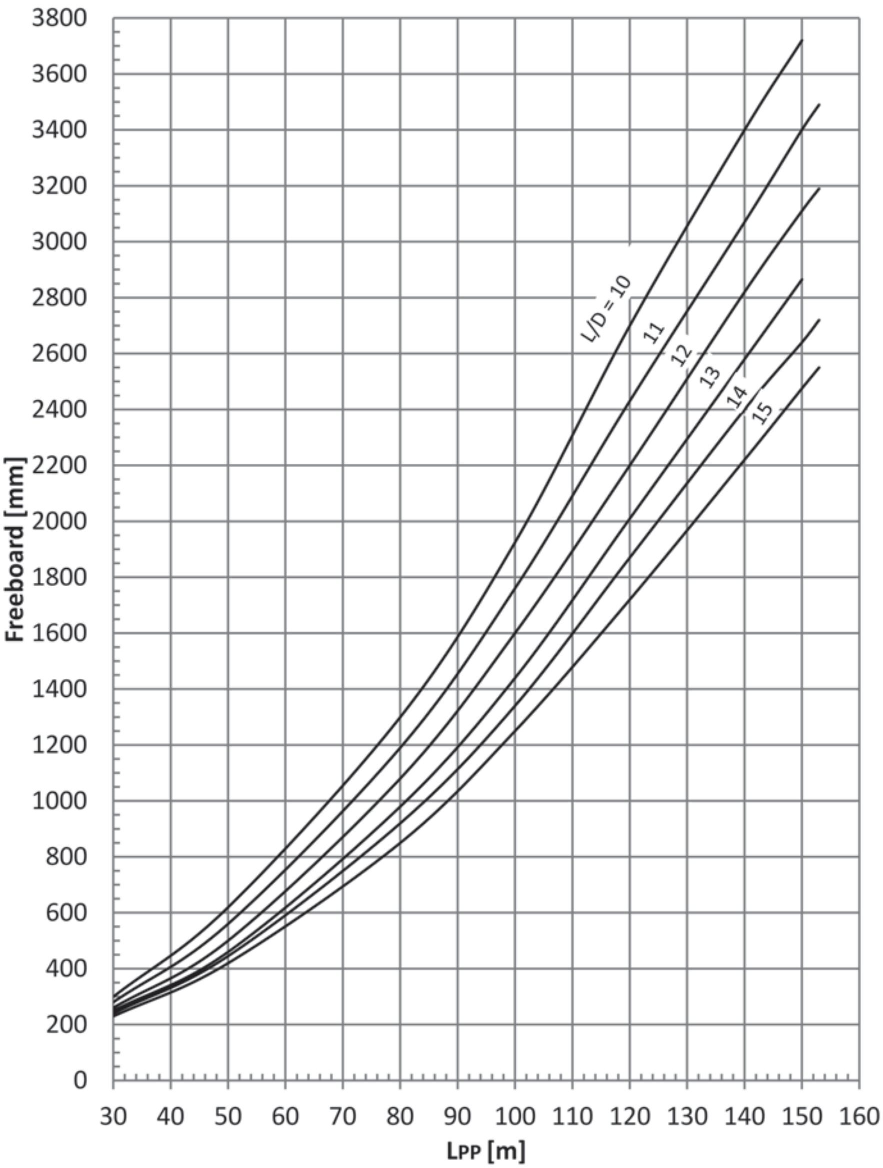


Fig. 2.104 Freeboard of dry cargo ships for $l_S = 0.1\ L_{PP}$

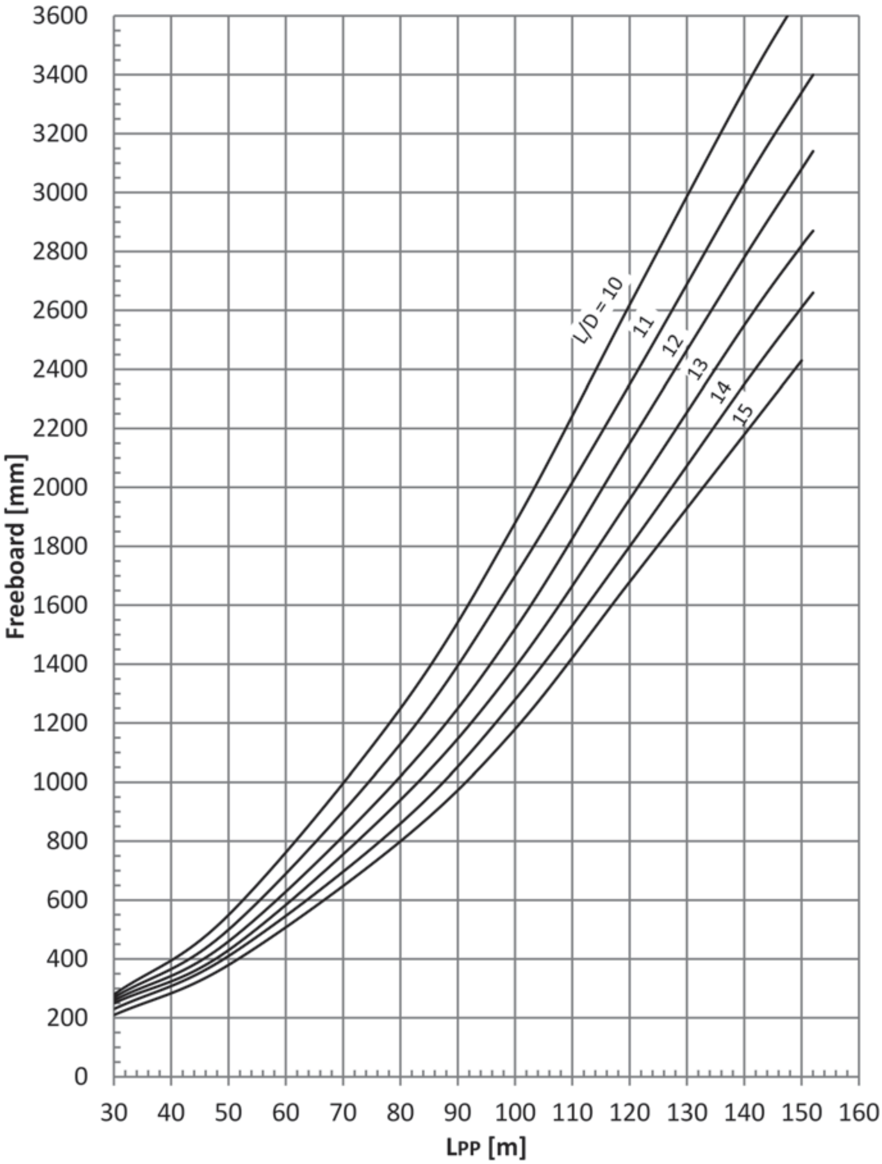


Fig. 2.105 Freeboard of dry cargo ships for $l_S = 0.2\ L_{PP}$

vessels, tugs, offshore supply vessels, etc., the existence of a sheer is absolutely necessary. The sheer also affects the ship's stability at large inclination angles and slightly the position of the floodable lengths' curve, as well as the resulting position of the watertight bulkheads, as they are required.

- *Load Line Regulations*: The ICLL Load Line Regulations (Reg. 38) specify for ships *without* or *reduced* sheer, in comparison to those with *normal* sheer, *in-*

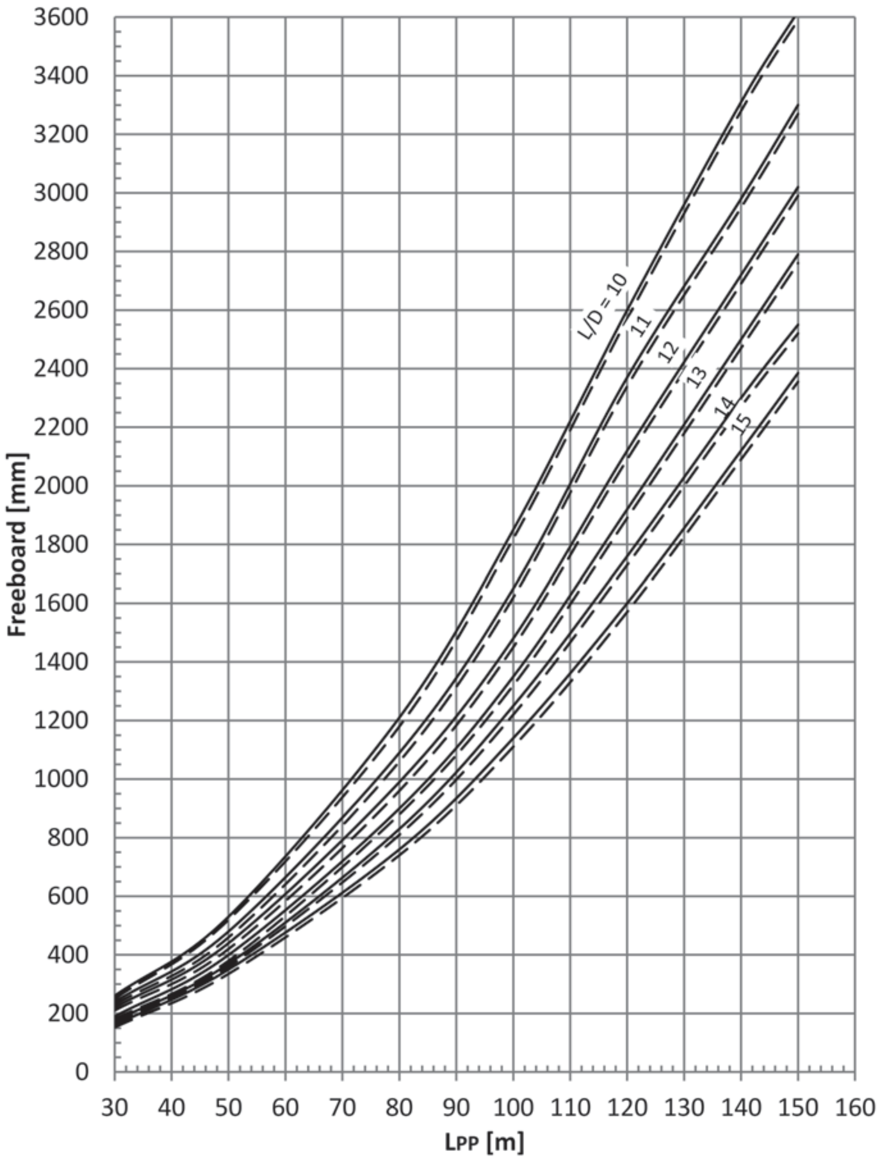


Fig. 2.106 Freeboard for dry cargo ships for $l_S = 0.3\ L_{PP}$. (———— ships with forecastle/poop, - - - - - - ships with forecastle, poop, and superstructure amidships ($l_{Smidship} > 0.2\ L_{PP}$)

creases in the required freeboard. In contrast, for *increased* sheer beyond the standard values, *reductions* of the freeboard are allowed. The standard/normal sheer is given by two parabolic parts, which extend to the forward and aft part of the ship. The focal point of the above parabola (zero sheer) is at amidships. The

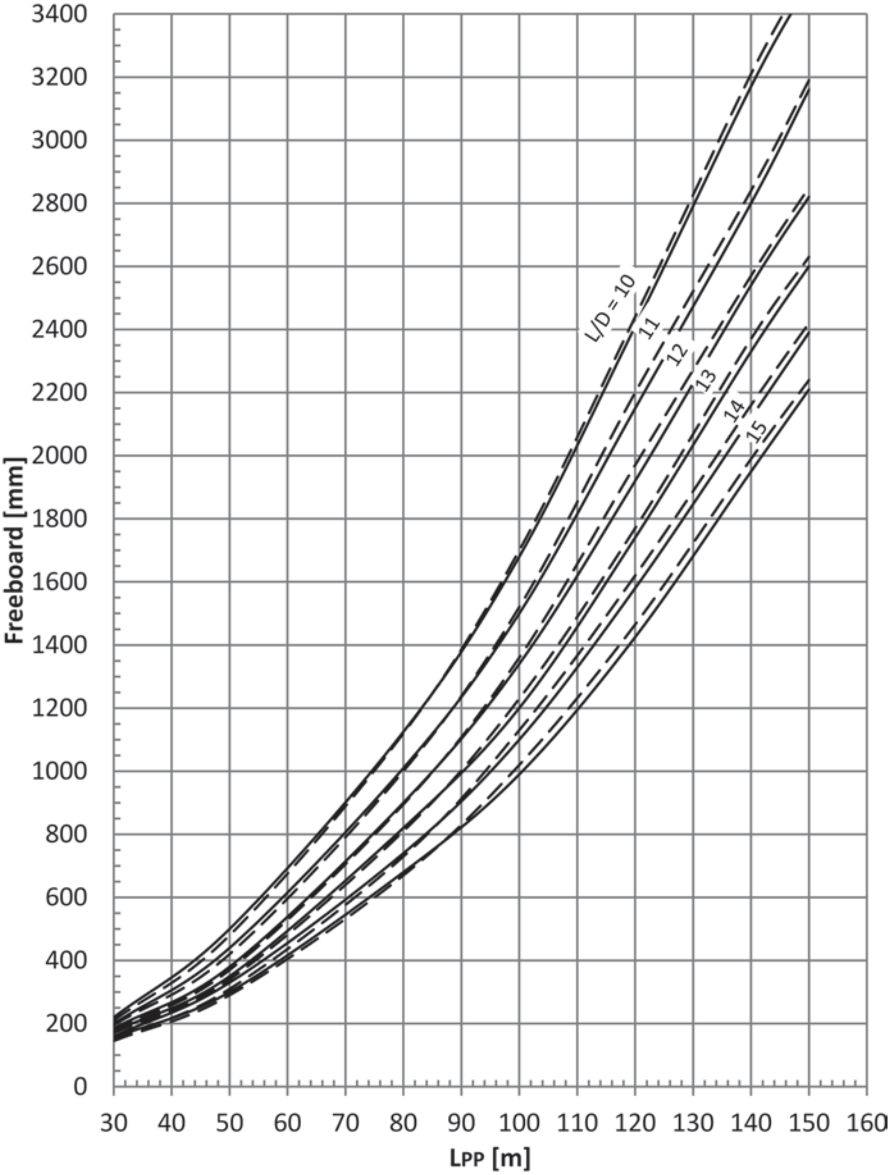


Fig. 2.107 Freeboard for dry cargo ships for $l_S = 0.4\ L_{PP}$. (———— ships with forecastle/poop, - - - - - - ships with forecastle, poop, and superstructure amidships ($l_{Smidship} > 0.2\ L_{PP}$)

normal aft sheer at AP is 50 % of the fore sheer at FP. The height of the standard fore sheer is:

$$S_A\,(\text{millimeter}) = 50 \cdot \left(\frac{L}{3} + 10\right)$$

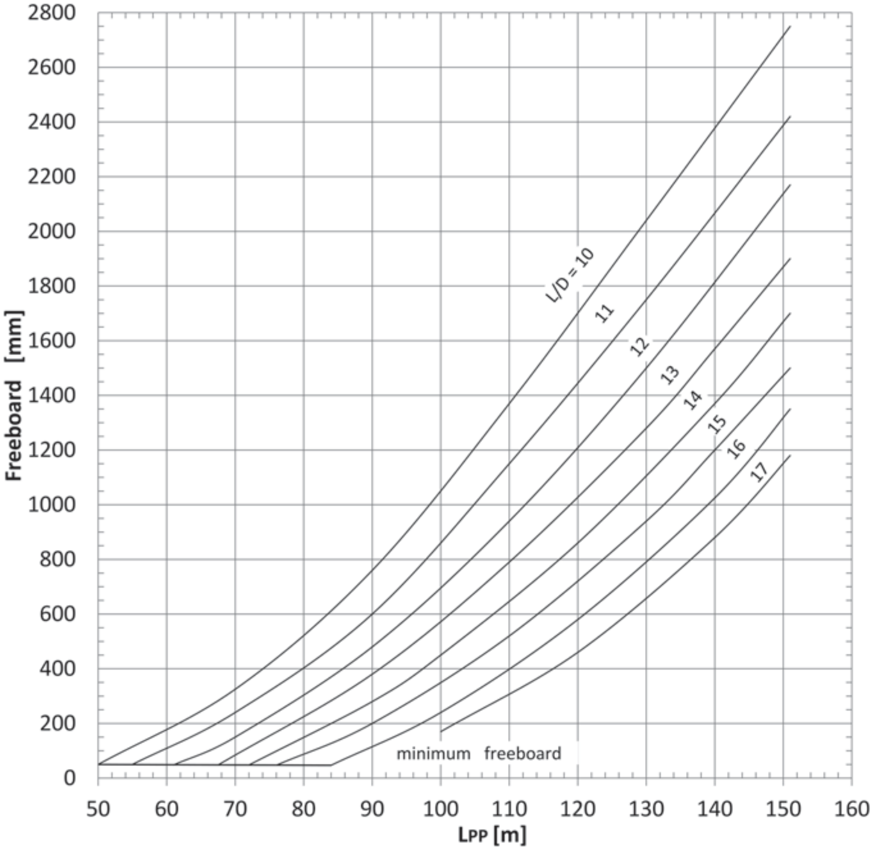


Fig. 2.108 Freeboard of dry cargo ship of type shelterdecker (ships with a protective deck) for $l_S = 0.99\ L_{PP}$

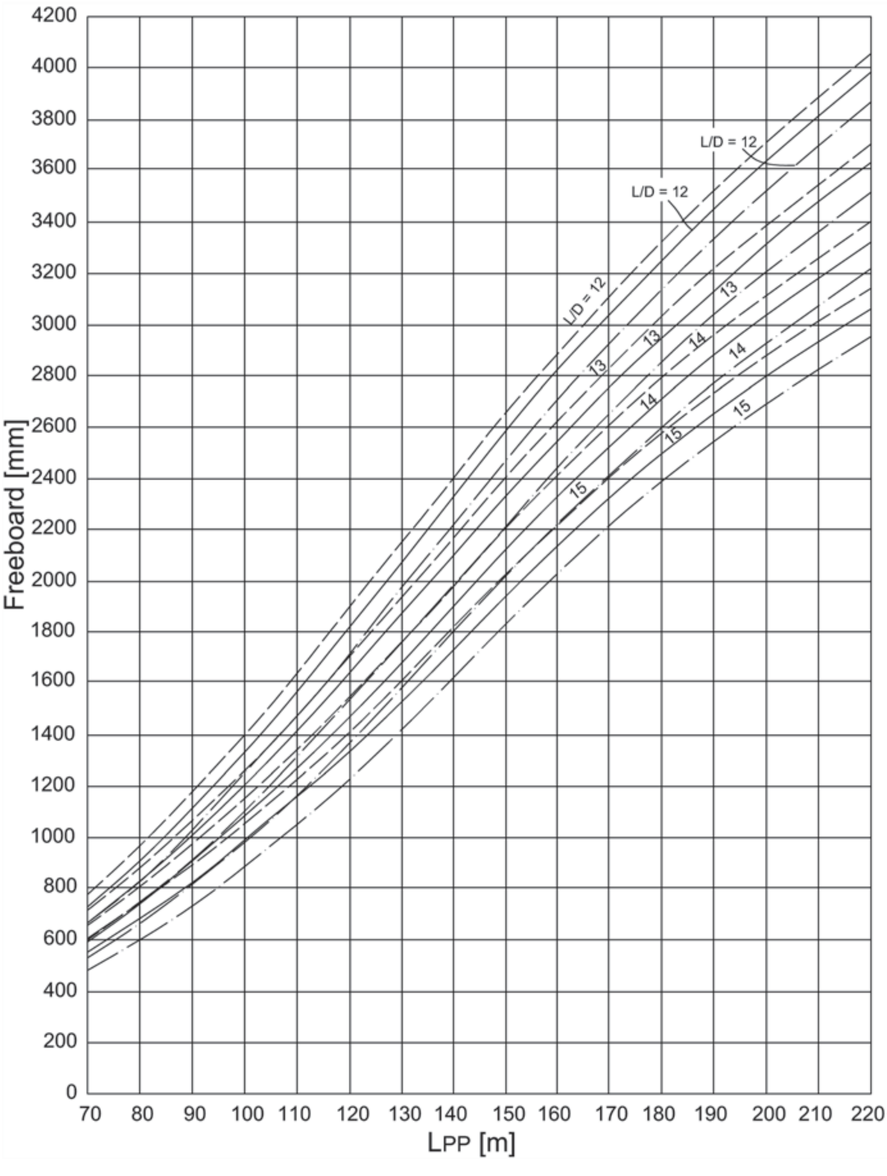


Fig. 2.109 Freeboard of tankers. (———— $l_S = 0.2\ L_{PP}$, --------------- $l_S = 0.3\ L_{PP}$, – – – – – – – $l_S = 0.4\ L_{PP}$)

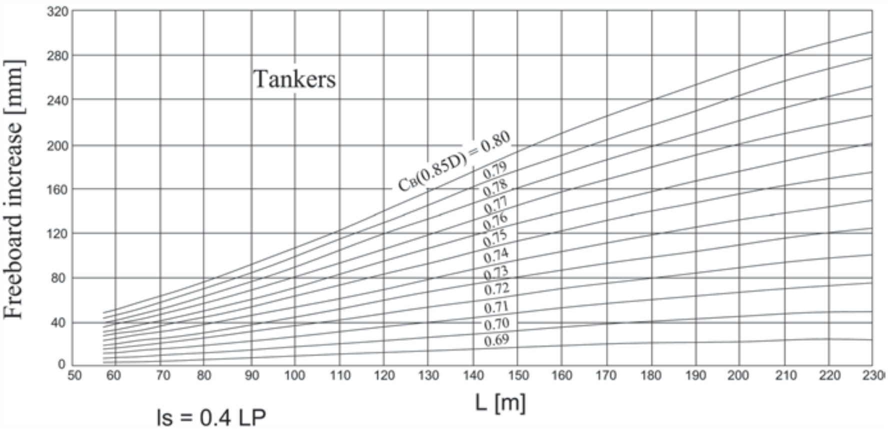


Fig. 2.110 Correction for C_B (0.85 *D*) ≠0.68. ($l_S = 0.4\ L_P$) (for C_B (0.85 *D*) ≤0.68 no correction. C_B (0.85 *D*)=∇0.85 *D*/*L*.·*B*·0.85 *D*, *D*=height of freeboard deck)

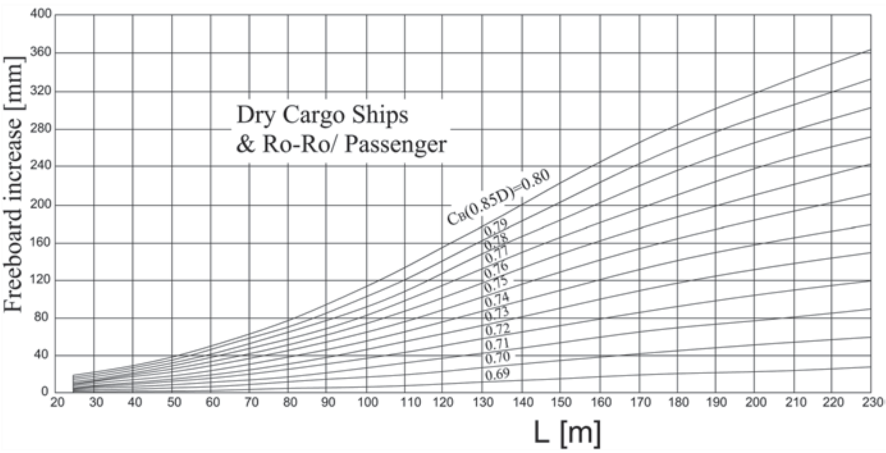


Fig. 2.111 Correction for C_B (0.85 *D*), no correction for C_B (0.85 *D*)≤0.68

where *L* (meter) is the length of freeboard calculation.

The following Fig. 2.112 presents typical sheer curves, regardless of the normal ones according to the Load Line Regulation, for various ship types. It is noted that for fast ships, the region of minimum sheer is abaft of the midship (~15–25% *L*); the same applies to tug boats. Also, the relationships for the fore and aft sheer height, as percentage of length and among themselves, vary according to the ship's speed and the requirements for adequate seakeeping behavior.

- *Deck sheer*: The sheer is measured at the side edge of each deck with respect to the waterline taken as basis for the calculation of freeboard. It should be noted,

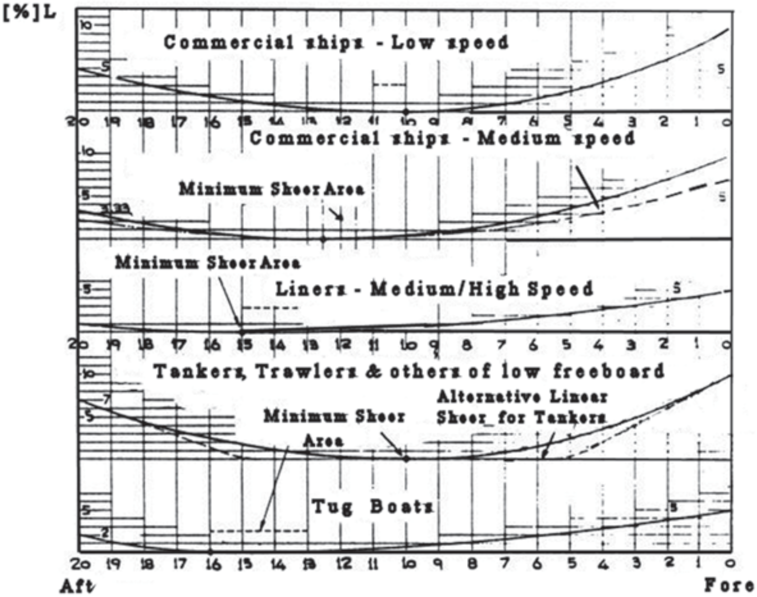


Fig. 2.112 Dimensionless sheer curves for various ship types

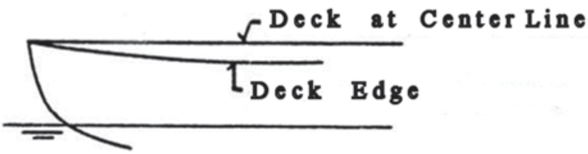
due to the existence of a camber typically across the ship's weather deck, that the resulting line of the deck at centerplane should be faired (see Fig. 2.113).

Uppermost decks exposed to weather (*weather-decks*) without sheer, but *with transverse camber,* have an even deck at the centerplane, while the deck line at the sides results from the height of the camber at the centerplane (usually $b \cong B/50$, where $B(x)$: breadth of reference deck).

The sheer of other decks except for the weather or freeboard deck, is obtained as follows:

- Decks *above* the weatherdeck, for example, superstructure decks, are usually constructed with the same sheer like the weatherdeck. However, on ships with intense sheer, for example, tugboats, fishing vessels, etc., these decks are constructed without sheer (at least the deck which accommodates the wheelhouse).
- Decks *below* the weatherdeck are generally constructed *without* sheer. This enables the exploitation of the additional stowage volume at the ends, and this is exempted from the tonnage of the ship. An exception here are passenger ships,

Fig. 2.113 Fairing of lines of deck with sheer and camber



when the second deck is their bulkhead deck, namely, the basis for calculating the floodable length. As is well known, the floodable lengths increase, if there is sheer, so it is not advisable to ignore the sheer in this case. Certainly, when it comes to car ferries, with zero sheer on the bulkhead deck (corresponds to the car deck), the distances between the watertight bulkheads are in anyway greatly reduced.

2.19.4 Critical Review of the Load Line Regulations (Abicht et al. 1974)

An analysis of the ICLL Regulations may be skipped in the context of the current textbook (see Antoniou and Perras 1984). However, regarding the effect of these regulations on the design of a ship, the following is noted:

1. The relationship of the required freeboard to the ship's length specifies for small ships not only absolutely, but also as a percentage of length, small freeboards (see Fig. 2.103, percentage basic freeboard for ships of type B). This appears to be contrary to the principle of ensuring sufficient buoyancy and seaworthiness for all ships independently of their size, while allowing smaller boats to operate in relatively heavier seas (on the master's responsibility). It should be noted, however, that this is directly related to the level of operational risk of the ship sailing in normal and severe environmental conditions; from the point of view of regulations, it is generally accepted that a larger ship should be safer than a smaller one. Consequently, the risk levels should be reduced when increasing the size of the ship, such as when increasing the number of people on board (see new probabilistic regulations on damage stability, SOLAS 2009).
2. The specified relationship of freeboard with a series of technical characteristics of the ship, for instance, the ship's type (A or B), size, superstructures' extent and sheer, does not always reflect the ship's actual safety requirements, which would result from a first principles study (seakeeping calculations) and correlation of the above parameters in a rational/scientific way.
3. The required survivability level, in case of damage, for large tankers (type A ships, $L \geq 150$ m), although logical, does not fit to the general context of the ICLL regulatory framework, nor explains the exclusion, from similar requirements regarding the watertight subdivision, of other risky ships, for example, small short-sea cargo ships.
4. Generally, the international regulations on load lines (ICLL) and stability after damage (SOLAS) should be harmonized into a unified regulatory framework. Relevant consultations among the working group committees of the International Maritime Organization IMO have not yet led to practical results.

Despite these critical points, taking into account the recent amendments of the currently in force ICLL Regulations, it is considered that the safety of in-service ships is satisfactorily covered by existing regulations. However, the appropriate imple-

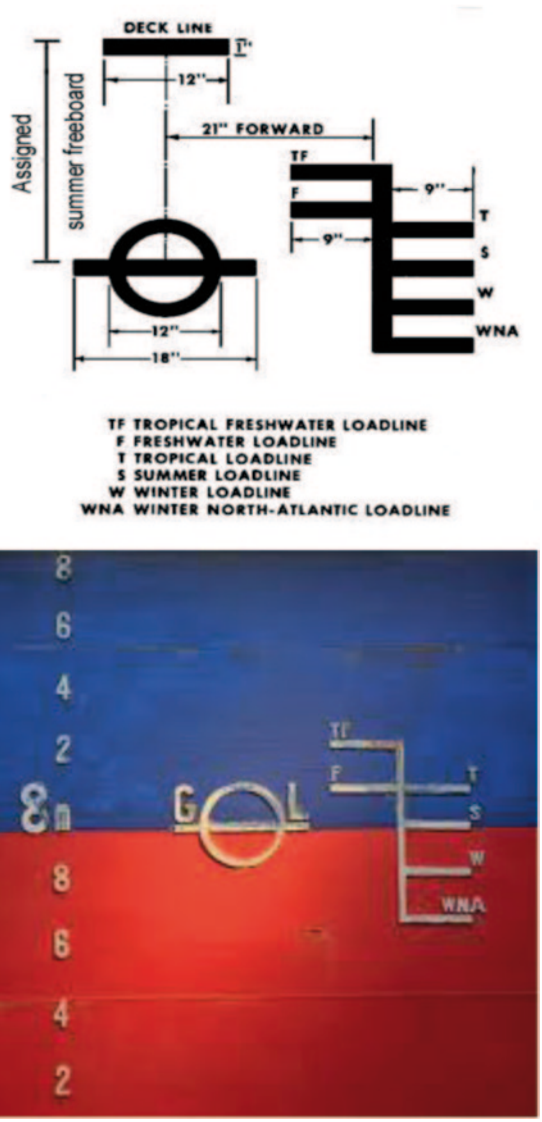


Fig. 2.114 Load Line Mark (Plimsoll Line)

mentation of the load line regulations in practice (control of the "actual" freeboard, *Plimsoll's* mark, Fig. 2.114) relies on the reliability of the various inspection bodies (local port/coast guard authorities); bad ship operation and improper inspections may lead occasionally to disastrous consequences (accidents from overloads of ships).

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